Naval Architecture as Art and Science

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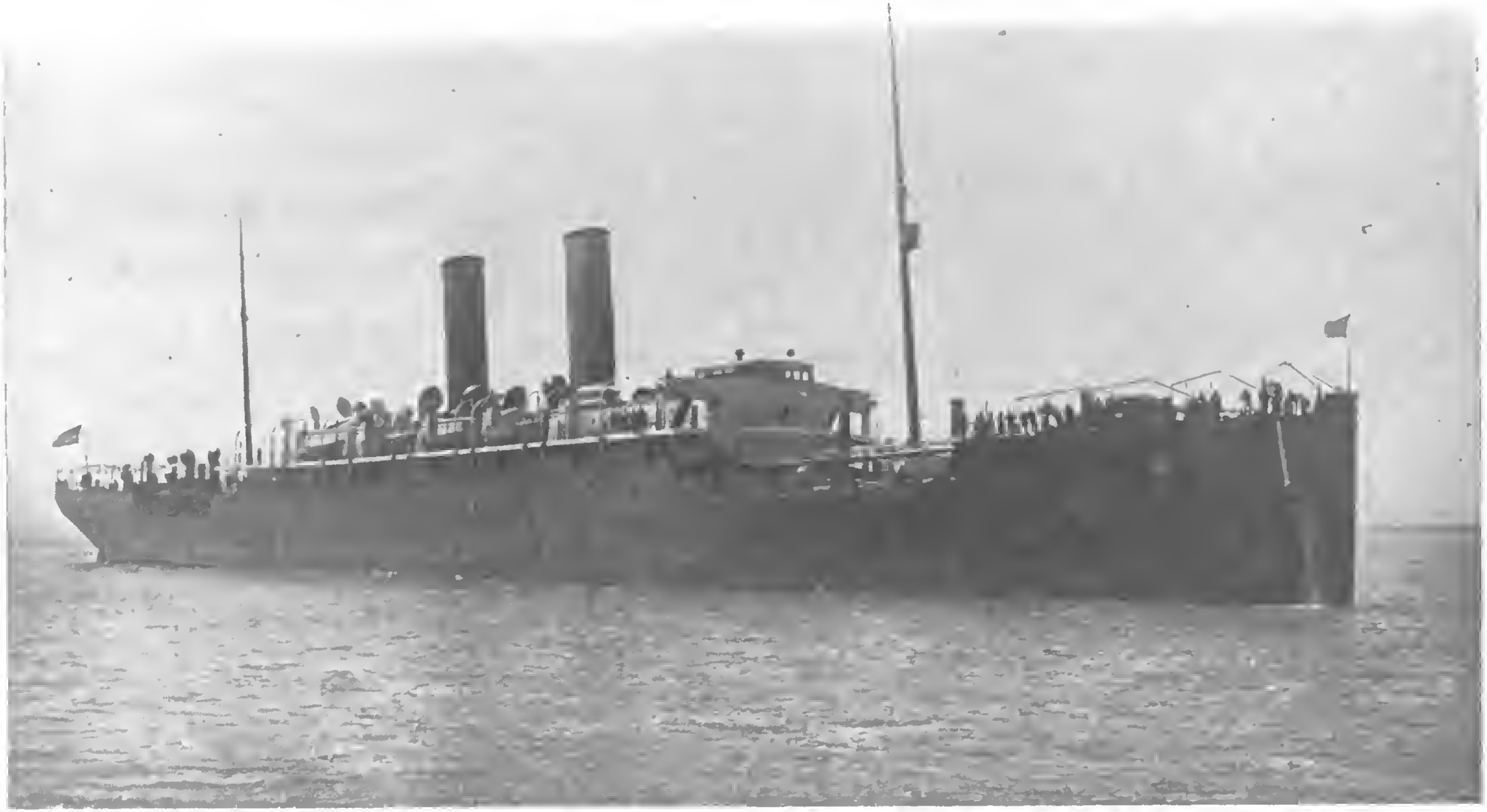

S/S Korea, Pacific Mail Steamship.

Built 1899 by the Newport News Shipbuilding Company, and designed by the author on his sinoid system for highest speed. (18 knots estimated, 21 knots actual, with 8280 tons deadweight.)

Naval Architecture

as Art and Science

by

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Naval Architect

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To My Wife

Foreword

Any ship-minded person who has seen the splendid specimens of Viking Ships still preserved must ask: How could such shipbuilder's art degenerate into the formless, clumsy ships of the Dark Ages?— And the answer must be: Lack of definite guideposts. Indeed, not until the conception of the Baltimore Clipper, in 1825, can it be said that the old guideposts were rediscovered and made useful. This book has been written with the purpose of elevating the old guideposts into scientific *lodestars* for permanent improvement of the ship designer's art until it has reached the status of an exact science. A big task—how performed, the reader must judge for himself!

This book had its beginnings sixty years ago when the author, studying Dixon Kemp's *Yacht Architecture*, began to see how guesswork dominated the designing of ships. This opinion was subsequently strengthened during the author's college years; but it was not until 1930 that his efforts took a definite direction. The cause was a paper *Frictional Resistance of Ship Models* read by Lieutenant W. P. Roop, U.S.N., before the Society of Naval Architects in New York. This paper opened my eyes to the inadequacy of the present Froude-Taylor theories of resistance. Since ship designing centers more or less on resistance, this inadequacy was fatal. However, with the assistance of model experimenters all over the world, the author has been able to develop new theories and new formulas for resistance culminating in the Differential Calculus method of solving the problem of ships' Optimum form, proportions, and dimensions, whether for investment as in a merchant ship, or for efficiency as in a warship or a yacht. It should be distinctly noted that the new theories are developed from and agree exactly with the most accurate model experiments ever made in the United States, in England, in Austria, and in Germany. To refute the theories would be to refute the experiments.

Another great help in my work was the munificent gift of the Transactions of the Institution of Naval Architects (London) from 1869. This gift by Mr. Wm. G. Starkweather of Boston enabled the author to go to the source of all our present knowledge of ship designing and building.

Many new subjects have been treated in the book, such as the most favorable longitudinal lines, bow and stern angles, form of minimum wetted surface, effect of basin walls on model resistance, river towboats, propeller immersion, optimum speed-length ratios, and differential derivatives for minimum resistance per ton of displacement. The proposal for a new criterion of wave heights in strength calculations is urgently brought to the attention of the Ship Register Societies.

Acknowledgments

The Author is greatly indebted to the U. S. Bureau of Construction and Repair (now Bureau of Ships) and the Experimental Model Basin, Navy Department, Washington, D. C., for their never-failing information on difficult questions in model experiments. Special mention must be given Captain William MacEntee, Captain H. R. Saunders, and Captain L. B. McBride, who with unwearying patience answered innumerable questions and submitted many blueprints and other data.

Nor will the author forego to mention the great help given by Dr. G. Kempf of Hamburg Model Basin, and by the late Dr. K. Schaffran of Berlin Model Basin.

References indicated throughout the text by superior numbers will be found on pages 207 and 208.

C. O. L.

Contents

Part I

Elementary Principles

Naval Architecture is the art of creating, out of the caveman's dug-out, floating things of beauty like Clipper Ships or Yachts, or of usefulness like modern Cargo Ships, or of fighting power like Warships. Art means knowledge made efficient by skill, science is systematized knowledge changing from day to day, and exact science unchanging exact knowledge such as mathematics, geometry, and astronomy. Later in this book, Naval Architecture will be developed into an *exact* science, eliminating guess-work in ship designing.

A ship or vessel is first created in the designer's mind, then put on paper in a preliminary and a final form, and lastly built by men and machinery out of existing materials, according to the paper plans. Every one of these functions needs highly skilled men earning good wages or salaries. It is the purpose of this work to lead the novice through the various steps until he becomes a skilled man in his profession.

Fundamental Axiom. The total weight of a vessel equals exactly the weight of the displaced water hence it is called the displacement of the ship. This fact was first discovered by Archimedes, and is called the *Law of Archimedes*. It can be used to obtain the displacement of a model by putting the model in a tank full of water, and weighing the overflow instead of figuring the displacement from the plans of the ship—a rather intricate procedure.

FORM OF SHIPS

The general form of any vessel is a double wedge or spool with or without a middle body, the only flat part of the form, as a rule. Being curvilinear, it cannot, strictly speaking, be shown on a flat plan but this drawback is overcome by the expedient of cutting the model by plane surfaces in various ways, both vertically and horizontally. The vertical planes are called sections and correspond to the frames or ribs of the vessel. The horizontal planes are called water lines, the flotation plane being the load water line, another plane below is called the light water line. There are several other planes, to be mentioned later.

The longitudinal form of ships was probably first derived from fishes

hence the nickname *Cod's Head and Mackerel Tail* for vessels with a blunt forebody and a tapering long afterbody. This result was the outcome of model experiments by old Dutch shipbuilders who towed models at *the same speed* as the ship it represented. We know better now but it was not until the appearance of the famous racing yacht *America* in England, in 1851, that the "Cod and Mackerel" type was abandoned for the more evenly balanced form used today. Strangely enough, the old Viking Ships, built without theories or model experiments, possessed a perfectly balanced form, both ends being alike.

The criteria of good ship forms are low resistance to propulsion, high carrying capacity and seaworthiness. The ship designer must always strive to combine these three conflicting qualities in the most economical and efficient manner. As a thin wedge is easier to drive in, so it was held that the finer a ship's ends, the easier it should drive. For instance, Scott Russel,[1] the most experienced designer who ever lived, built a ship whose length was fourteen times its breadth. It was not found possible, on such proportions, to fulfill the conditions of highest carrying capacity, adequate strength, and low resistance and, at present, even the fastest naval vessels have a breadth to length proportion of 1/10, without any parallel middle body at all. Generally speaking, however, it has been found that the higher the speed in proportion to the length, the finer the ends must be shaped, but only at the expense of carrying capacity—paramount in merchant vessels.

The aim of the designer, then, is to find *the form, the proportions, and the dimensions* of a vessel, large or small, that shall fulfill the conditions in a given case, in the most satisfactory manner to its owner. Hitherto, the necessary choice has been purely imitative; originality is the rarest exception, as shown by the first Viking Ship, the first Clipper Ship, the first Monitor.

PROPORTIONS OF SHIPS

The ratios between the main dimensions are called the proportions of the ship. The main dimensions are length, breadth or beam, depth, and draft of water. As already explained, the ratio of breadth to length never exceeds 1/10, but at low speeds the best value of this ratio is $\frac{1}{3}$ if the keel be straight as in a merchant ship. With a draft to beam ratio of $\frac{1}{3}$, the smallest possible under-water surface per ton of displacement is obtained, a very important matter.

Draft is naturally the limiting dimension, and since only a few harbors permit a draft of 40 feet, the beam would be 120 feet, and the length 360 feet. Unfortunately, such dimensions would make the ship far too stiff, and cause heavy and violent rolling. But a smaller vessel could be given a length of 90 feet, a beam of 30 feet, and a draft of 10 feet without being considered extreme in any way, or a heavy roller. Such proportions are most satisfactory from the shipowner's standpoint, because the vessel would attain the highest carrying capacity and the lowest engine power per ton of displacement, in other words, the highest possible earning capacity, other things being equal. As length is increased, a parallel middle body is inserted to make up for the deficiencies in beam and draft. In this manner the earning capacity of a long, narrow ship is better preserved.

SHIP PLANS

As already mentioned, the curvilinear form of ships cannot be expanded on flat paper but the form is shown by imaginary cuts of the model in several directions. These cuts appear on the paper as curved or straight lines depending on the view or projection of the cut. The principal projections are the profile where the view is from the side, the half-breadth plan where the view is from above, and the body plan where the view is from the ends of the ship.

For the first ship plan of the student, the simplest ship form is chosen, a rectangular box with more or less wedge-shaped ends (*see* Fig. 1). The profile shows in this case a rectangle, the half-breadth plan a double, cigar-shaped wedge, and the body plan smaller rectangles. The profile is also called the sheer plan because it shows the sheer that is the rise of the deck line at the ends of a ship.

PRINCIPAL DIMENSIONS. The principal dimensions are of course length, breadth, and depth just as in any other body but there are several kinds of each dimension. Of lengths, length over all, length in the load water line, and length between perpendiculars; of breadths, beam at deck, beam at water line; of depths, molded depth, draft forward, draft aft, mean draft, freeboard forward, aft, and in the middle.

Length Over All (*LOA*) is the distance between the outermost parts of the ends.

Length in the Water Line (*LWL*) is the distance between the ends at the level of the load water line.

Length Between Perpendiculars (*LBP*) is in merchant ships, the distance from the forward end at the *LWL* to the after end of the rudder post to which the rudder is attached.

Beam at Deck (*B*) is the distance across the ship from side to side at deck, at the greatest breadth of the ship.

Beam at LWL (*BWL*) is the greatest distance from side to side in the *LWL*.

Molded Depth (*H*) is the vertical distance from the deck at side to the bottom of the ship, at mid-length.

Draft (*D*) is measured vertically from the *LWL* to the bottom of the ship whether forward, aft, or in the middle.

Freeboard (*F*) is the vertical distance from the *LWL* to the deck at side whether forward, aft, or in the middle.

Sheer (*S*) is the difference between the freeboard forward and in the middle (sheer forward), and between the freeboard aft and in the middle (sheer aft).

Freeboard and sheer are now regulated by *International Rules*, but it has taken thousands of years to do it.

Preliminary Plans. We are now ready for the preliminary plans of the ship. Suppose we want to draw the lines of the 90-foot ship mentioned above, $LOA = 90$ feet, $B = 30$ feet, $D = 10$ feet. Choose any convenient scale, say ¼ inch to the foot which means a length of 22.5 inches, a beam of 7.5 inches, and a draft of 2.5 inches on the plan. For clearness, the body plan is placed to the right of the profile or elevation. Adding *L* and *B* plus a 3-inch margin, we get 33 inches as the length of the paper. Adding *B* and *D* plus a 3-inch margin, makes the height of the paper 13 inches. The *title* to the plan is usually placed at the right-hand lower corner. We forgot to add the *F* to the height of the plan, say $F = 4$ feet which, brought down to scale, makes the paper 14 inches high.

Start now by drawing the *LWL* as a straight line, 2 inches from the upper edge of the paper, extending the line right across for the body plan to the right. Next draw the center line of the ship as a straight line 7.25 inches below *LWL*. Now draw a vertical line square to the *LWL* and 1 inch from the left side of the paper, extend it all up and down. Measure off *L* 22.5 inches along *LWL*, and erect another

vertical line. The two verticals are the perpendiculars, and in this plan, $LBP = LWL$.

Divide the distance between the *PP* into an equal number of parts, say six that represent the vertical cuts or sections of the model, and of these, three belong to the forebody of the ship, and three to the afterbody. The greatest or midship section is in this case midway between the *PP*, and is shown in its true shape in the body plan. For simplicity, a rectangular midship section is chosen and laid down on the paper to the right of the profile as mentioned above. Erect a vertical line midway between the fore *PP* and the edge of the paper which line represents the center line of the ship, also shown in the half-breadth plan. Complete the sides, the bottom, and the deck in the body plan by measuring off the beam, the draft, and the freeboard as already stated. Evidently, half the beam is put on each side of the center line in the body plan.

So far only straight lines have been drawn preliminary to depicting the curved form of the ship by the use of thin battens or splines weighted down to the plan by lead chunks called spline weights, or simply held by ordinary pins. Splines of uniform thickness held at three points on the plan, always form a curve of a certain shape called the *elastic line*. Because it ends and begins in straight lines, the elastic line is the most efficient curve for the longitudinal form of ships but it is not so good a sheer line that looks best with even curvature all the way like a circular arc. In order to draw the latter, the spline must be held at five or more points, two of them outside the ends of the curve—in this case outside of the *PP*.

With the flexible spline or batten held at three points, the greatest breadth at section 3 and at both *PP*'s in the center line, we can draw in the longitudinal shape of the deck in the half-breadth plan, which is also the shape of the *LWL* and all lines below because the sides of the ship are vertical and straight. With the usual ship forms, the deck line and all the water lines are different from each other although for obtaining the best hull efficiency, all these lines should have some similarity showing that they have been derived from a common origin. Next, use the batten for drawing the sheer line in the profile after the sheer, 2 feet forward and 1 foot aft, has been marked off on the forward and on the after *PP*. Bend and fix the batten so that it touches the spots on each *PP*, and is tangent to the freeboard line amidships.

Now transfer or lift the half-breadths at each section to the body

plan, and likewise all deck heights from the profile in order to obtain the outline of all sections in the body plan. Here the sections in the forebody and in the afterbody are alike except as to the sheer heights, while generally in ship plans, all sections have different outlines. Yet, as a matter of fact, the more similar the sections, the less the resistance, and the higher the efficiency of the hull. Sections and water lines have this property of similarity in common, no matter how different their outlines are. This fact has led many designers to attempt to derive the water lines geometrically—quite useless, since the flow of the water past the ship is never along the water lines but diagonally across the surface of the ship. In the body plan the flow lines always run square to the sections until the ship's bow waves disturb the even pattern of the flow. The flow lines expanded take the form of the previously mentioned elastic line, and are thus the principal means of shaping the hull. *Expanded* means here the entire flow line laid down flat on paper, as will be explained later. In passing, it should be noted that there can be no *streamlines* along vessels floating on the surface of water, nor along automobiles running on the surface of the earth, contrary to the general imagination. There are streamlines only in submarines at great depths, and always in airplanes and in dirigibles; but the elastic line is just as valuable, in air as in water, for shaping the longitudinal form.

DIAGONALS. There is another set of lines called diagonals because they cut diagonally along the body plan, making a certain angle with the center line. This angle is so chosen that it cuts most of the sections as near square to the surface as possible. Diagonals are excellent for fairing the hull surface, and should always be drawn first of all curved lines. In our case, the diagonals do not run square to the sections in the body plan, and are shown only for completeness.

BUTTOCKS. We have seen how the length is divided by sections, and the sheer plan by water lines, one of which is the load water line because the ship is supposed to sink down to this line when fully loaded for sea. In a similar manner, the half-breadth plan is divided by vertical planes called buttocks (*see* Fig. 1). These lines also serve the purpose of fairing the hull surface in conjunction with the water lines, the sections, and the diagonals. The lines are not faired until all crossings meet at corresponding places in the different projections on the plan. This locating of the crossings is sometimes very difficult and may cause serious alterations in the hull surface.

Our design is now ready for the lettering and the title, which should include the dimensions, displacement, cargo capacity, engine power or sail area, speed, and the scale of the plan, space for alterations, and other data.

Displacement. Our next job is to find the displacement of the ship from the design, which is very easy in this case, for we have first assumed the form and the dimensions, thus fixing the displacement beforehand. Since the sections in the body plan are made rectangular for simplicity, the displacement in cubic feet amounts to

$$\text{area of the water line} \times \text{draft}.$$

The area of any water line bounded by the elastic line is

$$\text{area} = 0.64 \times \text{length} \times \text{breadth} \tag{1}$$

Hence displacement $= 0.64 \times L \times B \times D$ in cubic feet, but is always given in long tons of 2,240 pounds. There are 35 cubic feet to one long ton, and

$$\text{displacement} = \frac{0.64 \times L \times B \times D}{35} \tag{2}$$

which formula is valid for any ship with rectangular sections, straight keel, and the water line bounded by an elastic line. Actually, the sections are seldom rectangular, and the coefficient 0.64 varies between wide limits of 0.45 and 0.95, its best value being 0.57 for a ship without parallel middle body. Clearly, the less the coefficient, the less the displacement for any given dimensions, but also, the less the carrying capacity of the ship, including fuel, water, stores, and crew, in addition to any cargo for merchant ships or any armament for warships.

Now tugs and yachts, for instance, carry no cargo and need a smaller coefficient than tankers and big cargo ships; but for reasons of resistance its value is seldom over 0.85. This coefficient is most important for the design of a ship, and is called *block coefficient* because it measures the ship's body fullness in relation to the circumscribed block or box.

Example of Displacement Calculations. Let us use the dimensions mentioned on page 6: $L = 90$ feet, $B = 30$, $D = 10$ feet, and block coefficient $= 0.64$. From formula (2)

$$\text{displacement} = \frac{0.64 \times 90 \times 30 \times 10}{35} = 494 \text{ tons.}$$

The displacement should be sufficient for all the weights that the vessel must carry; if not, the displacement must be increased by increasing the coefficient 0.64—*not* by increasing the main dimensions. For in the latter case, all the weights in the ship would be increased; whereas in the former case, only the hull weight would be slightly increased, and perhaps, in some instances, the machinery weights.

Coefficients and Their Symbols. In the design of this vessel, only two coefficients have been used, that of the area of the water line, and that of the displacement. But in ship designs of the usual form, many more coefficients are needed to show the main features, and it becomes necessary to use symbols for shortening the estimating work. We give here a list covering the hull design.

Cb block coefficient measures the fullness of the underwater body in relation to the circumscribed box.

Cp prismatic coefficient shows the relation between the ship's body and the circumscribed prism or cylinder based on the form of the midship section. *Cp* is sometimes called longitudinal coefficient, a name that could just as well be applied to *Cb*.

Cx midship section coefficient measures the area occupied by the midship section, of the circumscribed rectangle, the important equation

$$Cb = Cp \times Cx \tag{3}$$

proves that each of the three form coefficients is dependent on the other two.

Co load water line coefficient shows the area of the circumscribed rectangle occupied by the *LWL*.

Cl the lateral plane coefficient is a measure of the *ratio* between the lateral plane area and the circumscribed rectangle $L \times D$.

Symbol Examples. Of the vessel depicted in Fig. 1, $Cb = 0.64$, $Cp = 0.64$, $Cx = 1.00$, $Co = 0.64$, $Cl = 1.00$.

Centers. In order to make the ship float as designed, it is necessary to know not only the weights of the different parts but their centers as well. The principal centers are:

B the center of displacement or of buoyancy is the center of gravity of the hole made in the water by the ship's body. It is always located in the symmetry plane of the ship but its fore and aft and its vertical positions must be estimated from the design.

G the center of gravity of the entire ship must always be located vertically above or below B, generally above B. Only when the vessel is fitted with heavy ballast weights low down, can G be below B, as in sailing yachts with lead keels and no other ballast.

M the metacenter is likewise situated above B, and marks the upper limit of G beyond which the ship would either heel over or eventually capsize. Several vessels have been lost in quiet harbors from this cause.

The three centers B, G, and M are best shown on the body plan of the design. Note, however, that B leaves the symmetry plane when the ship is heeled over, whereas G and M remain in the plane. The distance MG is called the metacentric height explained in the section on STABILITY, pages 12–15.

PAPER PATTERNS. Paper patterns are a very handy means to obtain the center of any plane figure of whatever shape. Cut the paper to the contour of the figure, and balance it on a pinpoint, or hang the pattern with a short plumb-line from a pin stuck in the wall first from one end, and then from the other, as near as possible to right angles to the first one. Mark with a pencil where the plumb-line intersects the edges of the pattern in both positions, and draw pencil lines through the marks. The center is where the lines cross. The same procedure is used for obtaining the center of gravity of any volume, by cutting paper patterns of its sections (as in the body plan). By gluing the patterns together in their correct positions, the center can be found as in the case of a plane figure. Very little glue should be used, and only around the center of each pattern, in order not to disturb the pattern center position. In this manner, then, center of gravity of any volume is obtained with greater accuracy than by estimating from formulas.

CURVE OF AREAS. If the area of each section in Fig. 1 or any ship plan is laid off as a straight line in the half-breadth plan, and a curve is drawn to the spots on each section, it is called the Curve of Areas. In other words, the areas are treated as straight lines of certain length, laid off to any convenient scale to keep the Curve of Areas within the

confines of the half-breadth plan. In this way the volume of the ship is represented as a plane figure, the center of which coincides with the center of gravity of the volume in a fore-and-aft position, and can be found most easily by paper patterns as described. The curve of areas also gives an inkling of the resistance of the ship, hence it is a very important characteristic of the design. Unlike the longitudinal lines that go to form the body of the vessel, which are constant in shape, the curve of areas becomes very different in outline for a straight keel, and for a curved keel. In general, it cannot be represented by a geometrical curve.

Rudder. For steering a ship in a given direction, a rudder is needed, although one is not shown in Fig. 1. It is always placed as far aft as possible in order to gain leverage between the center of the rudder area and the pivoting point of the ship. This point is never at the center of gravity *G* of the ship, but near the stem when the ship is in ahead motion, and near the stern-post when in astern motion, in which case the rudder is very inefficient. In some vessels where high turning ability is desired, a bow rudder is fitted for going astern.

Problem. The reader should now draw all lines shown in Fig. 1 on a separate paper to a scale of ¼ inch to the foot. Thus the *LBP* of the 90-foot ship becomes 22.5 inches on paper, and all other dimensions in proportion. For the curve of areas, the sectional areas should be drawn to a scale of 1/32 inch to the square foot in order not to coincide with the *LWL* curve.

STABILITY

Unless a vessel can float right side up under all conditions of loading and of ocean surface, it is useless for the transport of cargo or armaments. Curiously enough, many ships have capsized in sheltered harbors although fully able to cross oceans when loaded to capacity. With light cargoes such as cork, a cargo vessel cannot be loaded to capacity, and might lack stability, whereas a heavy cargo like steel plates would sink a ship filled with it, or make the ship too stiff if loaded near the bottom of the vessel. Stability is thus dependent on the nature of the cargo, but fundamentally on the ratio of beam to draft as well as on the form below water. A round barrel or log, for example, has no in-

herent stability, but rolls over at the slightest touch. A square barrel or log possesses great inherent stability if its beam is more than twice its draft, in which case its equilibrium is said to be *stable* because the log always tends to return to its upright position if heeled or inclined to either side. If, however, the beam is less than twice the draft, the log cannot remain upright but tends to turn on its side or its bottom—a condition called *unstable* equilibrium. The round barrel or log is said

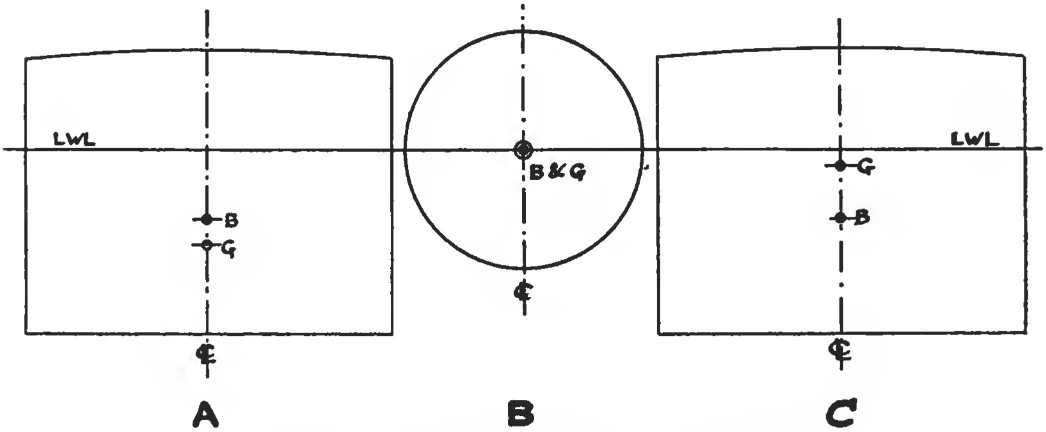


FIG. 2. MIDSECTION SHOWING CENTRA STATIONARY.

to have *neutral* equilibrium because it will remain upright in any position if made of homogeneous material. But any of these cases of equilibrium can be altered by introduction of ballast, whether water or heavier matter, in the bottom of the vessel. The effect of ballast is to lower the center of gravity G in relation to the center of displacement B as well as to the metacenter M (*see* page 11). If G is below B as in Fig. 2A, the equilibrium is always stable no matter how much the vessel is inclined. If G coincides with B as in Fig. 2B, the ship is in neutral equilibrium but still stable, and remains stable until G has reached M, when the vessel is liable to capsize unless assisted by sponsons or bulges near the water line.

ANGLES OF HEEL. The inclination of a ship is measured in degrees from the upright position (*see* Fig. 3). There are 90° between the vertical and the horizontal, and these 90° are called a right angle. There are four right angles or quadrants (360°) to the full sweep of a circle. The angle of inclination in Fig. 3 is only a few degrees, enough to show how the center of displacement B moves toward the immersed side to B_1. The opposite side is termed the emerged side. Between the upright *LWL* and the inclined *LWL*, two wedges are formed—the im-

mersed and the emerged wedge. The former acts as a lifting force, the latter as a downward pull, both of the same magnitude. Since nothing has been added to the displacement, it is clear that both wedges also must have the *same volume.*

If a vertical line is extended from B_1 to the center line CL of the ship, it meets the latter at the point M, the metacenter. If M is above G, the ship is always stable; but if the inclination is increased, there

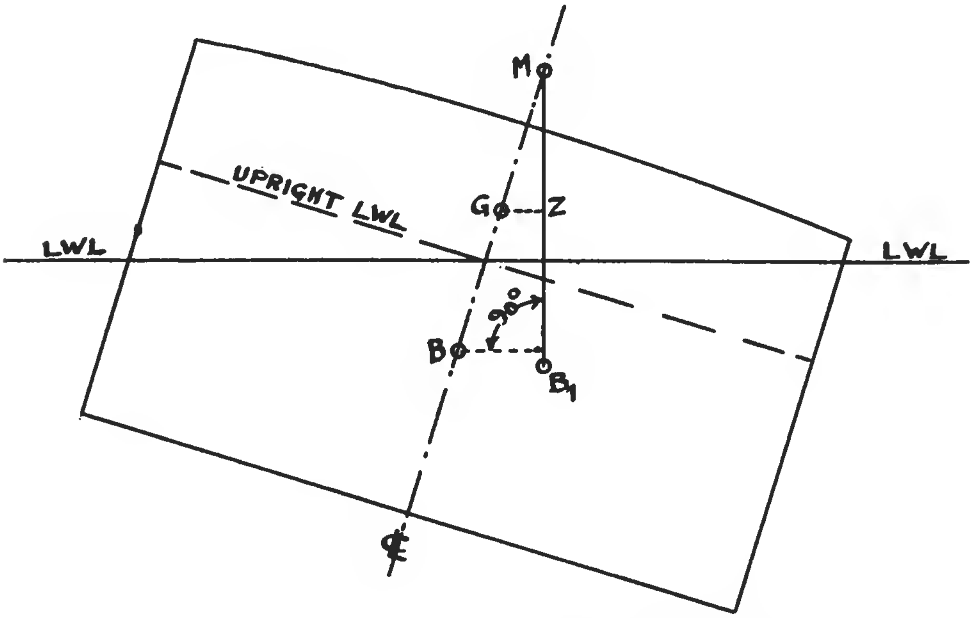


FIG. 3. MIDSECTION SHOWING CENTRA MOVED TO ONE SIDE.

comes a moment when M falls below G, and the ship capsizes. M is generally in a constant position for the first 10° or 15°, but a ship is mostly stable for 75° or three-quarters of a right angle, unless water gets into the hold. This *free water* not only increases the heel but becomes a source of danger if the ship is rolling in a seaway, and is a serious problem for the ship designer.

It should be noted here that the angle of inclination is outlined in Fig. 3 by BMB_1, or the angle between the CL of the ship and the vertical through B_1, and is the same as the angle between the upright and the inclined LWL's.

The stability here treated is termed transverse stability because the ship is heeled in a transverse direction. If in a fore-and-aft direction, the stability is called longitudinal.

Stability is measured by the *moment of stability* that the ship pos-

sesses on account of its weight, its form, and its dimensions. A moment is made up by a weight acting at the end of an arm or lever. For instance, if the weight is 10 pounds and its lever is 2 feet long, the moment is 2×10 or 20 foot-pounds. In Fig. 3 the weight is the displacement in tons of 2,240 pounds, and the lever is GZ, square to the upright or vertical B_1M. If G coincides with B, then the moment of stability would equal the moment of the wedges, the immersed wedge acting with an upward force, the other with a downward pull as mentioned above, their lever being the distance of the wedge centers. This fact makes it possible to estimate the position of M in any ship, as will be shown in Part II, pages 62–5. Generally, B is different from G, and the wedge moment different from the moment of stability, except at very small angles of heel.

RESISTANCE

A friend of mine, when resistance was mentioned, stated that a ship had no resistance in water—"You know how easily one can pull a boat to a pier." He did not know that the resistance of a ship can reach millions of pounds at high speeds; but such mistakes are all too common among laymen.

Its resistance makes a ship unique among structures built by man, insofar as its shape must be suited to the lowest possible resistance *per ton of displacement.* Where other structures need only strength at lowest cost, a ship needs strength, low resistance, seaworthiness, and low cost combined. Low resistance was early recognized as a *sine qua non* of a vessel—we need only think of the Viking ship—but the art was lost in the darkness of the Middle Ages. Not until the advent of the American Clipper ships was low resistance again in its place in ship designing, although the Swedish Admiral af Chapman had an inkling of it in the 17th century. He developed a theory for the longitudinal lines of a ship, which he assumed to be parabolas (curves very like the elastic lines mentioned in the earlier part of this text). He has been called the first Naval Architect, an honor that he richly deserves.

However, it was not until William Froude discovered the component parts of resistance that any progress in ship designing as an art could be expected. It should be distinctly noted that it is not low resistance *per se* that is required of a satisfactory design, but the lowest resistance per ton of displacement at the speed wanted. For example, an 8-oared

racing shell makes very little resistance even at 12 knots but its resistance per ton of displacement is enormous, and it cannot carry any cargo at all except its crew.

Froude discovered the component parts of resistance about 75 years ago, and his ideas have persisted to this day, although the errors have become more and more evident. He gave as component parts, skin frictional resistance, wave resistance, and head and eddy resistance, an unnecessary complication since all but the frictional resistance depend on pressure differences along the surface of the ship in contrast to the skin friction that is caused by the *adhesion* of the water to the surface of the ship. More pertinent names are *frictional* and *pressure* resistance, and will be used here.

But Froude's principal claim to fame is his rediscovery of Newton's model laws of the 16th century called by Froude the *Law of Similitude.* It says that ships of similar form and proportions but different lengths, and hence different displacements, are acted upon by pressure resistances proportional to their displacements if their speeds are proportional to the square roots of their lengths.

Unfortunately, ships can never be exactly similar in practice (*see* page 5) because of limitations of draft and stability, and—although similarity is fundamental in the model and its ship—unless the component parts of resistance can be separated, the Law of Similitude is of no use. From a towed model, only the total resistance can be measured, of course; but if one of the components can be estimated beforehand, the other is obtained by subtraction.

So Froude set about to compose a formula for the frictional resistance by towing planes of different lengths, painted or covered with different substances, and *fully submerged.* He obtained some values for lengths between 2 feet and 50 feet of the planes, but only by manipulation could he get his formula to agree with his experiments. The planes being fully submerged made the result of little value to surface ships, and he could not allow for the effect of the curved surfaces of ships. Still, his experiments were a great step forward in our knowledge of ship resistance. So far as is known, he was the first to conduct such experiments in water on a grand scale.

When applied to ships over 50 feet long, the Froude and all similar formulas gave values of skin resistance that were 50% too low for a 1,000-foot ship at a speed of 30 knots. This over-estimating of pressure resistance naturally resulted in the lean, unprofitable and inefficient

vessels of today, saved from utter uselessness only by superior materials and by more economical engines.

Two other factors unknown in Froude's days complicated greatly the subject of resistance. One is the Laminar Flow discovered by Reynolds in 1883 but not applied to ships until much later. At low speeds, below 3 knots for models, the flow of water particles reduces the skin resistance appreciably. The other factor is called tank wall effect, which very much increases the pressure resistance, in some cases nearly 50%. Both factors will be treated fully later.

WETTED SURFACE. It has been mentioned that the resistance reacts on the ship through its skin or wetted surface, and like other surfaces is measured by its length and its breadth multiplied by a constant dependent on the contour of the surface. In a submerged vessel, all its surfaces are wetted; but in a surface ship, only the sides and the bottom count as wetted. Take our design in Fig. 1, for instance: The area is measured by twice its draft times its length for the sides, and the bottom area by its beam times its length times the constant 0.64 (*see* page 9).

$$\begin{aligned} \text{Wetted Surface, sides, } 2 \times 10 \times 90 &= 1{,}800 \text{ sq. ft.} \\ \text{bottom } 30 \times 90 \times 0.64 &= \underline{1{,}730} \\ \text{Total W.S.} &= 3{,}530 \text{ sq. ft. (approx.)} \end{aligned}$$

This value is correct for estimating frictional resistance but, for painting, a small percentage must be added for the curvature of the sides, making a total of 3,550 square feet. In actual ships, the difference between the wetted and the painted surfaces is larger.

The surface can also be measured by its wetted girth, in our case twice the draft plus the beam, and by its length multiplied by another constant 0.785, found from experience,

$$\text{Wetted surface, } 50 \times 90 \times 0.785 = 3{,}530 \text{ sq. ft. (approx.)}$$

We can now proceed to estimate the frictional resistance of our design in Fig. 1. The formula is, *wetted surface times the square of the speed times a constant*, 0.0068. Suppose we want a speed of 10 knots, which is quite high for a 90-foot ship. The square of 10 (symbol 10^2) equals 100, and

$$\text{frictional resistance} = 3{,}530 \times 100 \times 0.0068 = 2{,}400 \text{ lbs.}$$

Expressed in symbols, the formula becomes

$$R_f = WS \times V^2 \times 0.0068 \qquad (4)$$

R_f = Frictional resistance in pounds
WS = Wetted surface in square feet
V = Speed of ship in knots.

The constant 0.0068 presupposes smooth, newly-painted steel surfaces of ships about 100 feet long, but it is much higher for models on account of the slightly sticky property of water and most fluids, termed viscosity.

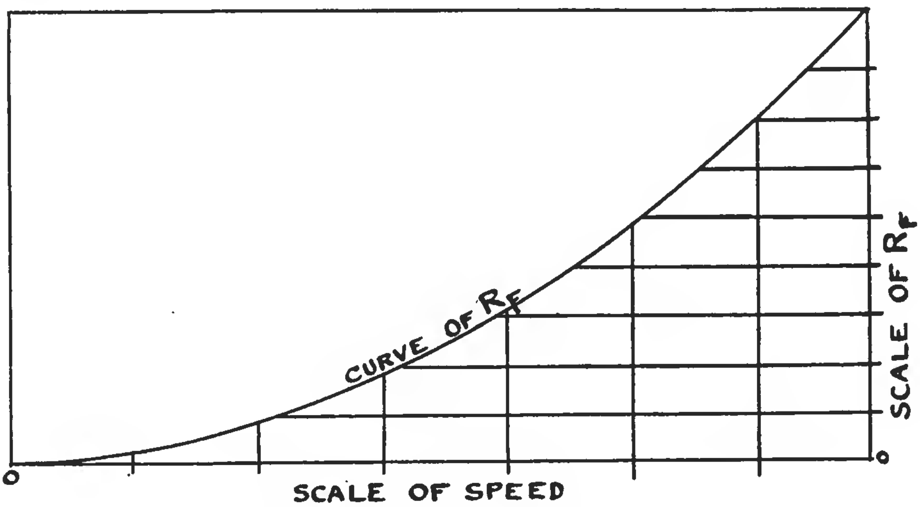


FIG. 4. CURVE OF FRICTIONAL RESISTANCE.

Like most formulas, (4) can be visualized in a diagram as in Fig. 4. On a straight line as a base, the speeds in knots are marked off to a certain scale, say ½ inch to the knot, and a vertical line is drawn at each knot so marked. Choose another scale to represent pounds, say ½ inch to 100 pounds. On each vertical line, mark off the resistance in pounds to scale, beginning with 10 knots and 2,400 pounds (*see* above). The resistance varies as the square of the speed, hence at 5 knots it is ¼ of 2,400, or 600 pounds, at 2.5 knots, the resistance is ¼ of 600, or 150 pounds, and so on. The curve drawn through these spots is called a Parabola of the second order because it varies as the square of the speed (V^2). If it had varied as the cube of the speed, $V \times V \times V$ or V^3, it would be termed a Parabola of the third order, and so forth. Parabolas are very common in diagrams of ship designs.

PROBLEM. Draw to scale a diagram of Fig. 4 as outlined here for the

90-foot design, and for another twice as long, 180 feet. For similar designs, the wetted surface varies a length squared, and the displacement as the cube of the length, or as L^2 and L^3 respectively. Estimate by formula (4) the frictional resistance of the 180-foot vessel up to 14 knots.

The pressure resistance will be considered in Parts II and III, as its nature is too intricate for beginners. Here it can only be stated that the pressure resistance always runs in a series of humps and hollows when visualized in a diagram, but that the mean curve varies as the fourth power of the speed, or as V^4, termed a Parabola of the fourth order, when the pressure resistance is corrected for errors in the estimates of skin resistance, and for the wall effect. The former correction is important at *low* model speeds, the latter at *high* model speeds, thus making corrections necessary at all speeds.

WEIGHTS OF SHIPS

A ship is always designed to carry a certain load aside from its own weight; hence it is important to know the latter weight, which includes the hull, the machinery, the outfit, and the fuel. A preliminary estimate is based on percentages of the displacement, varying with the size, speed, and uses of the ship as, for instance, a cargo ship, a passenger ship, a tug, a warship, or a yacht. For a vessel like the one in Fig. 1, displacement 494 tons (page 10), weights should average as follows:

Item	*Weight in Tons*	*Percent*
Hull, steel and iron	198	40.0
Wood and outfit	60	12.0
Machinery, Diesel engine	136	28.0
Useful load, fuel, water, stores	100	20.0
Total	494	100.0

For a big modern cargo ship, welded construction, driven by steam turbines, displacement 13,900 tons, the weights are:

Item	*Weight in Tons*	*Percent*
Hull steel, welded	3610	26.0
Wood and outfit	769	5.5
Machinery	736	5.4
Fuel, water, stores	1640	11.8
Cargo	7145	51.3
Total	13,900	100.0

In the first example, the vessel is supposed to carry no cargo at all except what is needed for its own use. In the second example, the ship carries over 50% of its weight in cargo, hence the greatly varying percentages. Neither a warship nor a yacht is supposed to carry any cargo except for its own use, but can always do it in an emergency.

STRENGTH

Every ship needs strength of three kinds, longitudinal, transverse, and local. If a plank is laid on two supports, and people sit down on it, it bends or deflexes, and the heavier the load, the stronger the plank must be made. A ship in the trough of two waves is subjected to such deflection. If the plank is laid on only one support in the middle as in a seesaw, its two ends bend down, and this is exactly what happens to a ship on the crest of a long wave. The two planks are examples of longitudinal bending. Four sticks nailed together to form a frame can have no strength whatever unless two corners are nailed with several nails or otherwise supported to prevent distortion. The frame is made stronger if all four corners are rigid. This is an example of transverse strength, provided for in a ship by the bulkheads that subdivide the ship into several rooms or holds, and by corner stiffeners termed brackets or knees. Local strength is needed wherever there is a concentration of heavy weights, as in the engine and boiler rooms, or where ocean waves buffet the ship, as at the bow.

All weights, loads, or pressures cause stresses that have to be borne by the ship's structure the principal parts of which are the outside skin (deck and shell) and the inside framework required to stiffen the thin shell. Up to a hundred years ago, wood was the only material used in a ship's structure, but now steel is used exclusively except in boats and small vessels. Sometimes the deck of coasting vessels is made of wood but it has little strength value compared to steel of equal weight.

The outside skin is built up by a number of plates riveted or welded watertight together, forming a continuous covering of the inside framework. The latter consists of frames and floor plates that stiffen the shell, and beams that stiffen the decks. The ship's bottom has to carry most of the load, and is therefore strengthened by the floor plates, which are really very wide frames. Most modern ships are required to carry water ballast and are fitted with watertight inner bottoms that add much to the safety of a ship. In some passenger ships this inner

bottom is carried along the sides up to the load water line. The feature was made compulsory after the *Titanic* disaster of 1912 with a loss of 1490 lives, the most far-reaching disaster in shipping history.

Besides the bending of the hull structure like a plank seat, termed *sagging*, and like a seesaw, termed *hogging*, there are immense stresses arising from grounding or docking, and from the ship's rolling in big ocean waves, which the structure must be able to withstand. Complicated stresses cannot hope to be solved by theory alone, hence all thicknesses of plates, angles, etc., have been fixed by trial and error. Only in regard to hogging and sagging stresses has a theory been developed that agrees fairly well with practice. It has been found that the load causing bending is very closely given by length multiplied by displacement and divided by the coefficient 35

$$\text{Bending Moment} = \frac{L \times D}{35} \tag{5}$$

For our design in Fig. 1

$$\text{Bending Moment} = \frac{90 \times 494}{35} = 1270 \text{ foot-tons.}$$

The plating in the sides, bottom and deck must be made thick enough to stand the resulting strains without breaking. How the difficult computation is made, will be shown later. There are technical societies, called Registries, that have made computations of materials for every conceivable vessel up to 600 feet in length, and from their Rules all dimensions of the materials built into a ship are obtained.

SUBDIVISION

It has already been mentioned that bulkheads provide a part of the transverse strength of a ship, and that the bulkheads also subdivide the ship into compartments. In an engine-driven ship, there must be at least four bulkheads, one at each end of the ship, and one at each end of the machinery space. But most large ships are given many extra bulkheads in order that the ship will float with one or more compartments flooded. There are now very strict International Regulations for the spacing of the bulkheads of passenger vessels carrying twelve or more passengers based on estimates of their *floodable lengths*, that is, the safe length of every compartment. (*See* page 79.)

LAUNCHING

For a ship built on dry land, some means must be provided to get her into the water. Her great weight makes it dangerous to move her in any direction but seawards. The means are termed *launching ways*, on which slides the *launching cradle*, in the bosom of which the ship rests when launched. The ways are permanent or standing, very wide wooden rails strongly supported by piling or concrete walls, given a certain inclination and extended far out into the water if the ship is large. The lower part of the cradle rests and slides on the standing ways, for which purpose grease is smeared thickly on the ways before the cradle is built, shortly before the launch. The upper part of the cradle is neatly fitted to the shell plating as far as possible fore and aft, because the cradle has to carry the full weight of the ship. At low water, just before launching, the seaward part of the ways is greased, the ship of course being launched at high water.

Launching is made both endwise and sidewise, the latter only when the opposite shore is too close. In any case it is a very ticklish business, as the ship has to be lifted clear of the building or keel blocks by hundreds of wood wedges driven into the lower part of the cradle, and at the same time kept from sliding down beforehand by dog-shores or triggers fastened to both cradle and ways. There have been many launching failures, the most notable, perhaps, the launching of the giant *Great Eastern* in 1857. The ship was launched sideways, and stopped on the ways, from which it cost a fortune to remove her.

Ships can also be brought into their element without launching if built in a dry dock. When the ship is ready, water is simply let into the dock and the ship floated out; it is a very costly mode of building.

After the launch, the ship is hauled into the fitting-out berth of the builders, its engine and other fittings put in, and the ship made ready for trial trips and delivery.

TONNAGE

There are many kinds of tonnage in use in the shipping business, which are confusing to the beginner. A cargo or merchant ship is measured by *Deadweight Tonnage*, expressing the number of tons of 2,240 pounds that a vessel can transport in cargo, stores, and bunker fuel. It is the difference in displacements between the *light* and *load* water line *LWL*, the latter marked by International Freeboard Statutes.

For settling of tonnage and harbor dues, the vessel's interior spaces not open to the weather are measured in units of 100 cubic feet called *Gross Tonnage* (G.T.). A ship of 1,000 G.T. thus has interior spaces of 100,000 cubic feet. From this G.T. certain deductions are made for space occupied by the propelling machinery, by fuel, by navigation and by crew spaces. The result is the *Net Tonnage* (N.T.), on which all dues are paid, and expresses the space available for passengers, mail and cargo. Both G.T. and N.T. are practically the same in every maritime nation.

The tonnage of warships is always the same as their normal displacement in tons of 35 cubic feet.

MATERIALS FOR SHIPBUILDING

As already stated, steel is now almost exclusively used in ships, except for cabin fittings that are now mostly made of fire-resisting materials. The many kinds of steel include mild, medium, hard, and nickel-alloy steel. For plates requiring flanges mild steel is used, but for most of the hull plates medium steel is preferred on account of its strength, and is also used in stern frames, stems, etc., that are cast. For extreme lightness coupled with strength, nickel alloy steel is used in big Atlantic liners, in warships of extreme speed, and generally in very fast vessels.

Cast iron is used for many articles in a ship's outfit, such as bollards, fairleads, and cleats, but cast steel is always used for important items, such as anchors. The difference between steel and cast iron is in the carbon contents, cast iron having about twice as much carbon as steel.

STEPS IN A CAREER AS NAVAL ARCHITECT

There are two ways leading to the top of the profession. One is to graduate from a technical college, which makes an applicant competent for the position of draftsman. The other way is to start as an apprentice in a shipyard, and to study textbooks on drafting and naval architecture in the evenings. In two years' time the apprentice should be able to start as a tracer in the drafting room, to rapidly advance to draftsman, then to leading draftsman, assistant chief draftsman, chief draftsman, assistant naval architect, and finally naval architect. A naval architect must have had at least fifteen years' training in a shipyard.

Part II

Advanced Studies

Jinia III, SHOWING BOW WAVE AND STERN WAKE. DESIGNED AND BUILT BY MATHIS SHIP BUILDING CO. SPEED-LENGTH RATIO, 1.59. *Courtesy of the Builders.*

THE STUDENT should familiarize himself with logarithms and with the slide rule, which save much time in repeated multiplication and division. He should have a working knowledge of algebra, geometry, and trigonometry, as well as an understanding of the planimeter for measuring areas on a drawing. The slide rule and the planimeter save an enormous amount of time in ship designing. They belong with the tools of the modern ship draftsman just as do pencils, ink, erasers, drawing instruments, scales, curves, splines, spline weights, straight edges, T-squares, triangles, thumb tacks, drawing boards, etc. Some of the items are supplied by the shipyards, and if the draftsman starts to buy the others gradually, the cost need not be excessive.

The student should further know the meaning of simple equations such as $y = Cx$, $y = Cx + Kz$, exponential equations $y = Cx^n$ and the universal equation $y = Cx^n + C$, the most useful one in the whole realm of ship designing. Above all, he needs knowledge of how to compute areas, volumes, and weights, most of which are taught in geometry lessons. Generally, the curved or curvilinear shape of ship areas, etc., make special methods necessary.

MEASUREMENT OF AREAS

Most areas in designing work have straight lines as bases, either horizontal as in water lines, or vertical as in sections. In many cases, the length of the area makes measurement by planimeter impossible or difficult—the water line of a 1,000-ft. ship is over 20 ft. in ¼ in. to the foot scale. Many means have been tried to measure such areas accurately, but only one is simple enough to be useful in ship calculations. It is called *Simpson's First Rule,* although really invented by James Stirling,[2] Scottish mathematician, in 1730.

Briefly, the rule says that the area is to be divided into a number of equal parts (2 or a multiple of 2) by straight lines perpendicular to the base, as in Fig. 5, where there are six equal parts, marked y_1, y_2, y_3, etc. The enclosing curve should be continuous—if not, the area must be divided into two or more parts. The curve between two and two of these parts (in pairs) is supposed to be a parabola of the second order, the simplest equation of which is of the form $y = Cx^2$. Any area of the

parabola bounded by perpendiculars equals $\frac{2}{3}\,hl$, h being its height, l its length. On this foundation the rule is built.

Next proceed as follows: Measure to scale the height of every ordinate y_1, y_2, etc., and multiply all odd numbers by 2, all even numbers by 4, and the two end-ordinates by 1. Add all the products together, and multiply the sum by one-third of the common distance between the ordinates. The result is the area of the figure in square feet, square inches, or whatever linear scale has been used. It is not even necessary to use the same scale for the base (length) as for the ordinates (height) if the right scale is used for each. Sometimes a long flat curve is contracted to one-half or one-third its length for reasons of testing its continuity or fairness.

Volumes are treated in the same manner as areas, the only difference being that the ordinates represent areas instead of heights, the areas first to be measured as explained above. Here it is seen that a planimeter would be a great time-saver.

EXAMPLE. In Fig. 5, page 29, measure the width or beam of the deck line at each section representing ordinates in Simpson's First Rule. y_1 and y_7 both $= 0$, y_2 and $y_6 = 7.3$ ft, y_3 and $y_5 = 12.7$ ft, and lastly $y_4 = 15.0$ ft. Arrange a table in similar fashion to Table I.

TABLE I

SIMPSON'S FIRST RULE FOR AREAS

(The common distance between ordinates is 15.0 ft.)

Ordinate Numbers	*Widths*	*Multipliers*	*Products*
1	0	1	0
2	7.3	4	29.2
3	12.7	2	25.4
4	15.0	4	60.0
5	12.7	2	25.4
6	7.3	4	29.2
7	0	1	0
		Sum	169.2

One-third of the common distance $= 5.0$ ft., $169.2 \times 5 = 846.0$ sq. ft.; but this is only one-half of the water line.

Water line total 1692.0 sq. ft.
Formula (1), page 9, area total 1728.0
Error 36.0 sq. ft.

This error amounts to 2%, due to a slight inaccuracy in drawing and scaling the elastic line used for the deck line.

Ordinarily, the error by Simpson's First Formula is much less except where the contour of the area is hollow or *concave* in some place, *S*-formed as it is termed, when closer spacing of the ordinates becomes necessary.

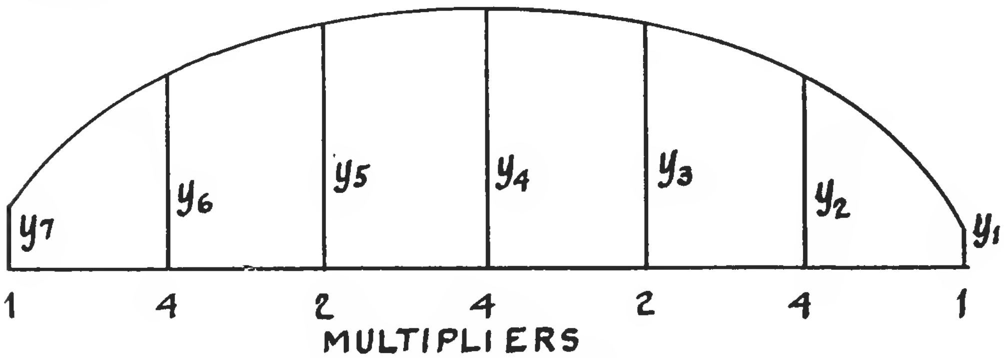


FIG. 5. ILLUSTRATING SIMPSON'S FIRST RULE.

FUNDAMENTALS OF SHIP DESIGNS

In the present state of the art, it is always necessary to make a preliminary design as a first approximation to the particulars of the required vessel—pure guesswork, with no assurance that the ship, if built, will be the best possible of its type. In Part III will be shown how the best dimensions, coefficients, and other particulars can be derived by higher mathematics from a primary design by simply altering the proportions. Unfortunately, limitations are imposed by stability requirements, by harbor depths, and by international regulations, so much so that the theoretical efficiency of 100% can only be attained by comparatively small vessels as sizes now go.

However, it has been stated in Part I, that the longitudinal form should be governed by elastic lines along the flow lines over the ship's surface. These flow lines show up as diagonals in the half-breadth plan, and as *trajectories* in the body plan, and must not be confused with streamlines of submarines and aircraft. (A trajectory is a curve that crosses a set of other curves at right angles.) The flow lines are distinctly new in ship designing, and a great step forward that shortens the work on the lines of a ship. For if its length in the *LWL*, and its midship section is known, we have only to select routes for the flow lines in the body plan to finish the sections with little or no fairing of the longitudinal lines, principally diagonals.

For instance, if the mid-section is half a circular arc, and the vessel were spoon-shaped, all sections would be smaller circular arcs, and the trajectories would all be radii. The width or radius of each section is determined by the elastic line drawn in the half-breadth plan. Evidently, the mid-section and the length being given, all the lines are fixed in advance. Mid-sections are seldom circular arcs, but in any case the flow lines must start at the greatest section and end at the center line of the ship.

The mid-section of a cargo ship is nearly a square because the square has got the shortest circumference for a given area. The corners at the bottom are always rounded, however. A fast passenger ship is given more rounded mid-section, yachts and very fast vessels, such as destroyers, get an almost elliptical mid-section that greatly eases the flow lines and thus reduces the resistance of the vessel at high speeds (*see* Fig. 6, A, B & C).

It may be argued that at high speeds the bow waves disturb the flow lines, but experience has shown that vessels designed in the manner here described, are less retarded by waves, even ocean waves, and hence faster. Hollow bow water lines are especially bad in this connection, creating double bow waves on each side of the ship, as shown by many photographs, and double wave resistance. Notable proofs of this truth are the modern racing yachts that never show hollow lines anywhere but whose speeds with their limited sail areas are amazing, indicating very small resistance.

With a given mid-section and length *LWL*, we must next select routes for the flow lines, and this needs some explanation. As the water is pushed aside by the bow, its tendency is to rise higher and higher as the speed increases. Vertical sides inhibit this tendency very little if at all, hence the sections near the bow should have vertical lines at the *LWL* and rounded below to prevent pounding in headseas. Such sections are termed *U*-shaped, and its flow lines are mostly horizontal.

After the water has passed the greatest section, the force of gravity makes it rush in from all directions, to fill the void aft of the vessel. To get the most out of this force, the sections in the afterbody or run should be similar in form to the mid-section; but as the keel of big ships must be made straight for docking purposes, a compromise results in what are termed *V*-sections, terminating farther aft in *S*-sections. The flow lines of these sections are more complicated than those of the forebody, but usually run from the bottom corner of the mid-section

to the *LWL* at the center line, always crossing the sections at right angles. Now a straight line is the shortest distance between two points, hence the flow lines should be straight in the body plan both forward and aft. In other words, the more parallel the sections run, the less resistance, as exemplified by the spoon-shaped vessel mentioned above.

William Froude was the first to prove, by the use of models, the lower resistance of *U*-shaped sections in forebody, and *V*-shaped sections in afterbody, and these sections have prevailed in ship designs

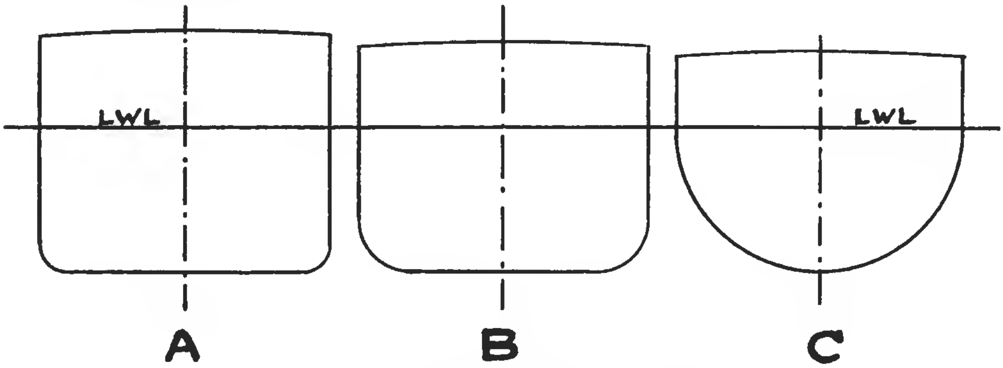


FIG. 6. DIFFERENT MIDSECTIONS.

to this very day. But Admiral Taylor showed by models in the Washington Experimental Model Basin that spoon-shaped afterbodies gave a still lower resistance—in other words, returned to the ship more of the gravity pressure on the afterbody when the ship is in motion. Very little is known of the Taylor experiment, the importance of which seems to have been entirely overlooked.

CRUISER STERN. A form of stern has been developed combining the advantages of the *V*-form, the *S*-form, and the spoon-form, and is termed a cruiser stern. It had long been in use in warships, especially fast cruisers (hence the name), but was not adopted in merchant ships until 1910 when *Prince Rupert*, Canadian high-speed coaster, was so constructed.[3] Such a stern combines a long *LWL* aft with fairly straight flow lines, and has a marked influence on the resistance. It is now almost universally adopted for both merchant ships and warships. All designs used as examples in this book are supposed to be fitted with cruiser sterns except where otherwise noted. It should be added that all racing yachts driven by sails have been built, for the last 100 years, with spoon-shaped after ends termed counters. No matter what the rating rule, the yacht type it produced was always built with a spoon-

shaped counter, a prolonged cruiser stern in fact. No better proof of its value is needed.

After this preliminary approach, we are now ready to begin actual designing of efficient vessels. Unlike structures on land, ship forms and proportions are decided mainly by the requirement that the resistance per ton of displacement must be minimum. For a given form and length of a ship, its resistance tends to increase with the speed; hence, form, length, and speed become primary considerations in the design. Form is decided by the mid-section and the longitudinal lines as already stated, so length and speed remain to be considered.

Speed-length Ratio. The combination of length and speed is not a direct ratio because, while the speed is taken in its unit knots, the square root of the length in its unit feet, is inserted in the speed-length ratio, thus

$$\frac{V}{\sqrt{L}} = C \tag{6}$$

If $V = 10$ knots, and $L = 100$ ft., $\sqrt{L} = 10.0$, and $C = 1.0$. Or, if $V = 14.14$ knots, and $L = 200$ ft., C again $= 1.0$. Ships that have the same C are said to have the same speed-length ratio, and if exactly similar, to have the same pressure resistance per ton of displacement (*see* the Law of Similitude, page 16). As explained, ships can never be similar in practice, hence the speed-length ratio, $\frac{V}{\sqrt{L}}$ would be of no value in ship designing except for two circumstances. One has to do with the model and the ship, which are always similar, and the other is connected with the *undular* or hump-and-hollow pressure resistance mentioned on page 19. A hump means an increase, a hollow a decrease in resistance compared to the mean; hence, a hollow should always be chosen for the speed-length ratio, but this is usually neglected by the ship designers. In all, there are only four such ratios, viz.

$$\frac{V}{\sqrt{L}} = 0.63, 0.72, 0.85, \text{and } 1.11 \tag{7}$$

representing a speed of 12.6, 14.4, 17.0, and 22.2 knots for a 400-ft. vessel. There is also another hollow at a speed-length ratio of 2.04, but this can only be reached by small craft, torpedo boats, and destroyers. For commercial craft, ratios below 0.63 and above 0.85 are

unprofitable. 1.11 sometimes is reached by very fast liners, but these are always subsidized by the State. Length has a secondary influence on the speed-length ratio, for while 15 knots may be useful for a 567-ft. ship, 6.3 knots are too low for a 100-ft. vessel, except tugs and inland towboats.

It should be noted here that the hollow-ratios in formula (7) are slightly different for the same ship and for vessels whose fineness and proportions are appropriate to their speed-length ratios. We are here thinking of displacement ships that do not plane and lift at very high speeds, such as hydroplanes that reach a speed-length ratio of 15.0 and over.

Curiously enough, although the pressure resistance per ton of displacement varies *inversely* with length, it will be seen later that frictional resistance per ton varies *directly* with length. The total resistance per ton of displacement is thus nearly constant within the same speed-length ratio, and increases very slightly with the length. This fact has a direct bearing on the weights of the machinery per ton of displacement.

EXAMPLE OF SHIP DESIGN

Besides the speed-length, there are many other ratios that must have attention when a vessel is designed, such as length divided by beam L/B, and beam divided by draft B/D. For a big ship with straight keel the latter ratio does not change very much from a mean value of 2.75 for efficient merchant ships, and is derived from the proportions for the minimum wetted surface, that in turn result in minimum frictional resistance and minimum hull weight. In contrast hereto, the L/B ratio varies within wide limits, 3.0 and 10.0. It is derived mainly from considerations of stability and metacentric height (*see* page 12 *ff*.). Formulas for all ratios are developed and proved in Part III, but at present we need only one,

$$B = CL^n \times \left(\frac{\sqrt{L}}{V}\right)^m \tag{8}$$

where C is a constant, n and m exponents, B and L as previously defined. For a $\frac{V}{\sqrt{L}} = 0.72$, a very serviceable ratio for cargo ships, $C = 0.89$, $n = m = 2/3$, and the last factor $= 1.245$. Hence $B = 0.89 \times$

$1.245 \times L^{2/3} = 1.11 \times L^{2/3}$. For instance, if we make $L = 300$ ft., $L^{2/3} = 45$ ft., and $B = 1.11 \times 45 = 50.0$ ft.

Assume now that our ship is to have a speed of 13 knots, and to carry a dead weight of 5,000 tons of 2,240 lbs. At $\frac{V}{\sqrt{L}} = 0.72$, $\sqrt{L} = 13/0.72 = 18.05$, and $L = 326$ ft., $B = 1.11 \times 47.3 = 52.5$ ft., $L/B = 6.21$. For this value of L/B the *curves of hull coefficients*, Fig. 29, page 132, reveal that the most efficient draft is 21.0 ft., 40% of B,

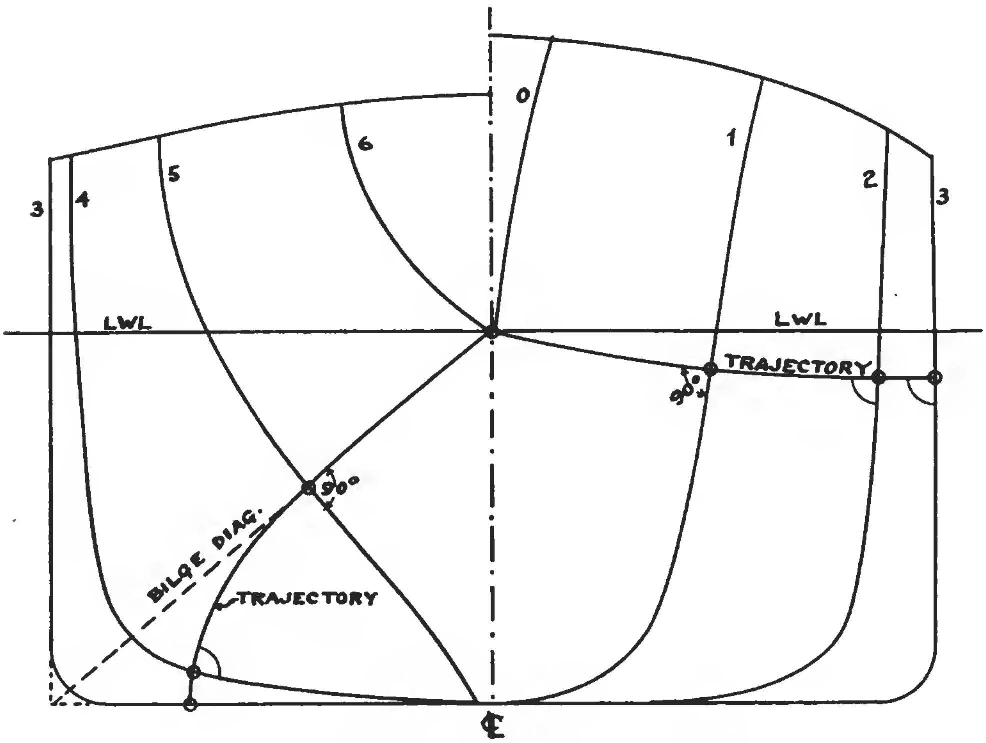


FIG. 7. BODY PLAN OF 5000-TON CARGO SHIP.

and B/D thus 2.50 instead of 2.75, the mean stated above. Since B is decided by stability requirements, and cannot be altered, the 21-ft. draft means 10% increase in earning capacity of the ship, and in the displacement. For the hull dimensions estimated, the resistance per ton of displacement is minimum, likewise the required shaft horsepower and machinery weights.

From the *curves of hull coefficients* we also obtain the prismatic coefficient $C_p = 0.70$, and as $C_x = 0.985$, the block coefficient $= 0.985 \times 0.70 = 0.688$, and

$$\text{displacement} = 0.688 \times \frac{326 \times 52.5 \times 21}{35} = 7070 \text{ tons (approx.)}$$

The required D.W., 5,000 tons, divided by displacement 7,070 tons, shows a dead-weight displacement ratio of 0.708—rather high, but it can be reached by an all-welded hull and by installing Diesel or high-pressure steam turbines of the latest types. Otherwise, a 3% increase in the principal dimensions will be necessary.

FREEBOARD. There remains now only the freeboard and the sheer to be computed from the International Load Line Regulations for Steamers. The freeboard for a length of 326 ft. is 51 ins., but is reduced by 23 ins. for a shelter deck above the freeboard deck if the height between the two is 7.5 feet. The molded depth $H = 21.0 + 2.33 + 7.5 = 30.83$ ft., the freeboard $F = 9.83$ ft. It should be stated that for a ship with a cruiser stern, the length in the regulations is 96% of *LWL*, or 312.5 ft., but *LWL* should be used in a first estimate to be on the safe side.

SHEER. Measured above the molded height line extended to the ends of the vessel. At the after end it is 10% of *LWL* + 10 ins., where *LWL* is considered in inches instead of feet, thus $32.6 + 10 = 42.6$ ins. $= 3.55$ ft. At the forward end it is twice as much, 7.10 ft.

LAYING OUT THE SHIP

The principal dimensions for the most efficient ship of 5,000 tons *D.W.* and 13 knots, are now estimated, and can be transferred to paper. In Fig. 7, the mid-section is first drawn to a scale of ¼ inch to the foot for greater accuracy, but the profile and the half-breadth plan can be drawn 1/8 inch to the foot in order to shorten this plan. Proceed now according to the directions for Fig. 1, and draw the body plan first—but the lower corners of the mid-section are rounded to a radius of 3.25 feet, not square as in Fig. 1. Next, the trajectories (*see* page 29) are located in the body plan. This is a simple matter in the forebody, where the trajectory would coincide with the *LWL* but for a slight forward overhang that makes it turn upwards at the center line. This overhang, at both ends, is useful in a head sea because it makes the bow rise faster to a wave. In the afterbody on the other hand, the

trajectory is not so easily located, but much help can be had by using the bilge diagonal for a first approximation.

Unlike Fig. 1, there is a long parallel middle body in this design, joining the forebody with the afterbody. The forebody is 98.5 ft. in length, the middle body 101.5 ft., the afterbody 126 ft., dimensions obtained from the aforesaid *curves of hull coefficients*. Notice that the afterbody is longest in order to get easy flow lines to the propeller.

Now divide the ends into three sections as in Fig. 1, the forebody spacings 32.83 ft., the afterbody spacings 42.0 ft., the middle body needs no sections, of course. In the completed design the sections are spaced 1/20 or 1/40 of the *LWL* for a more accurate estimate of the displacement and its center. In the forebody mark off a distance of $0.50 \times 26.25 = 13.13$ ft. from the center line outwards on section 1, and $0.866 \times 26.25 = 22.75$ ft. (approx.) on section 2. These are the ordinates of the flow line that ends on section 3, and should be laid off in the half-breadth plan too. A line drawn through the four spots shows the curve of the flow line or the main trajectory of the forebody, and intermediate sections can be measured or lifted for the body plan if necessary. Above and below the main trajectory the sections should be sketched in by hand, and faired by water lines, diagonals, or buttocks (page 8). Use Fig. 8 as a guide, it is the body plan of the primary design (page 29), the lines of which are exceedingly fair and harmonious. Use it for the afterbody too.

For obtaining the trajectories of the afterbody, we have to use the bilge diagonal that runs from the lower corner of the mid-section rectangle to the center line at *LWL*. The width of the diagonal is 32.30 ft., and for ordinate of section 4, 0.85 thereof is taken, $0.85 \times 32.30 = 27.50$ ft. (approx.). As ordinate of section 5, use $0.433 \times 32.30 = 14.0$ ft., both ordinates marked off on the bilge diagonal. Sketch in these two sections by hand, and fair with water lines and buttocks as in the forebody, until the route or *lay* of the trajectory is obtained. Measure its curved length in the body plan, and use the same factors for sections 4 and 5 as for sections 2 and 1. If the new ordinates agree with the positions of 4 and 5 on the trajectory, the lines are correct, otherwise the sections must be redrawn, beginning from the new ordinates.

The spots on the main trajectories have been marked by small circles in the body plan, Fig. 7. As a check to the correctness of the trajectories, the angles at their ends should be used, 23° forward and 19° aft.

In most plans of merchant ships, length B.P. is used for the design

length but it does not serve so well with a cruiser stern. Although the rules permit placing the after perpendiculars, A.P., 4% of *LWL* from the after end, this position would give too small a propeller aperture for a single screw ship.

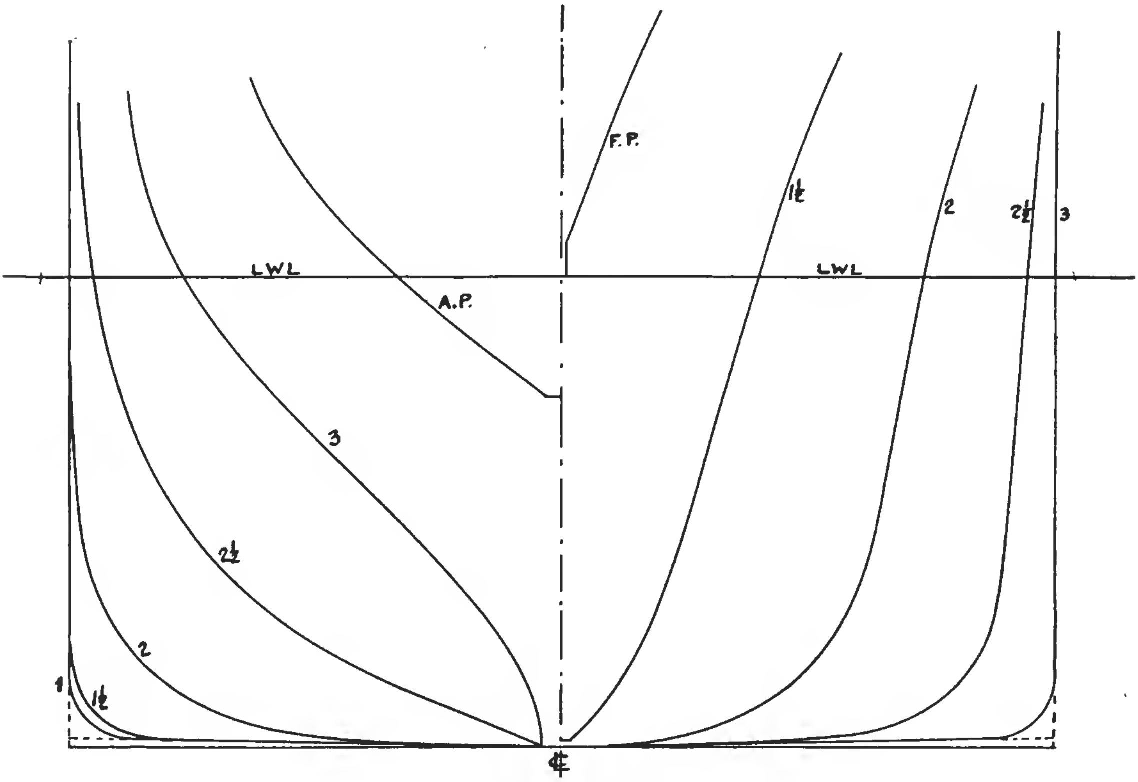


FIG. 8. THE PRIMARY BODY PLAN.

After the lines below the *LWL* have been faired, draw the overhangs fore and aft in the profile as well as the sheer line. For the shape of the overhangs look at some photographs of modern vessels, or use your judgment.

When the lines are all faired, divide the *LWL* in an equal number of parts, and compute the areas of the new sections and the displacement according to *Simpson's First Rule.* The longitudinal position of the center of displacement *B* is easily found if we remember that the moment of any area is equal to the sum of the moments of all its parts. As mentioned on page 28, volumes can be treated as areas if the ordinates are made up of areas instead of lengths, in this case the areas of the different sections (*see* Fig. 5). Arranged as in Table I, the sum of

the products multiplied by one-third of the common distance between the sections is very close to the actual displacement in cubic feet. The moment of each ordinate is of course its numerical value multiplied by its distance from the first ordinate. The moments can then be laid off as ordinates in a new curve and treated according to *Simpson's First Rule.* Note that the common ordinates in this case appear twice in the result that is squared. If the sum of the products is divided by the displacement in cu. ft., the result is the distance from the first ordinate of the center of displacement. The computation of the displacement and of the center of buoyancy can be made at one time.

TABLE II

SIMPSON'S FIRST RULE FOR DISPLACEMENT AND CENTERS

(Compare with Table I, the draft is 10 ft., the common distance 15.0 ft.)

Ordinate Numbers	*Areas*	*Multipliers*	*Products*	*Levers*	*Products*
1	0	1	0	0	0
2	73.0	4	292.0	1	292.0
3	127.0	2	254.0	2	508.0
4	150.0	4	600.0	3	1800.0
5	127.0	2	254.0	4	1016.0
6	73.0	4	292.0	5	1460.0
7	0	1	0	6	0
			Sum 1692.0		*Sum* 5076.0

$$\frac{\text{displacement}}{2} = 1692.0 \times 5 = 8460 \text{ cu. ft.}$$

$$\frac{\text{common distance squared}}{3} = \frac{15^2}{3} = 75.0,\ 75 \times 5{,}076 = 380{,}700$$

ft.^{2}-lbs.

$$\frac{\text{sum of moment products}}{\text{displacement}} = \frac{380{,}700}{8{,}460} = 45.0 \text{ ft. as the distance of}$$

center of buoyancy from the first section. The distance is exactly one half of *LWL* as it ought to be, both ends of the design being identical.

The student is now ready to make a similar estimate of the position of the center of buoyancy for Fig. 7. This center is very important in regard to the balance or trim of the loaded vessel, whose center of gravity must fall exactly in a vertical line from the center of buoyancy. For this reason, it is always best to cut a paper pattern of the curve

of areas, and to hang it up with a plumb line in two places. The center of buoyancy is where the two plumb lines cross.

ENGINE POWER

In order to be sure that the ship, when ready, will float at the desired trim, it is necessary to estimate the engine power to drive the vessel at the required speed at sea, and then from power and type, estimate the weight of the complete machinery. With the old steam engines, power was measured by indicated horsepower termed *IHP*. One horsepower, as defined by James Watt, the inventor of the condensing steam engine, equals the work done by lifting 33,000 lbs. 1 ft. in one minute. The 33,000 lbs. represent the resistance, and 1 ft. per minute is the speed at which the resistance is moved. At sea the unit of speed is not a foot per minute but a knot per hour, which is equal to 6,080 ft. per hour. One foot per minute means 60 ft. per hour—almost 100 times less than 1 knot, 101.33 to be exact. Hence when knots are used, the 33,000 lbs. must be divided by 101.33 = 325.67 lbs. With these units, the horsepower is expended in overcoming the resistance of the ship R at a speed V in knots,

$$EHP = \frac{RV}{325.67} = RV \times 0.00307 \qquad (9)$$

This formula shows the effective horsepower *EHP*, and R is the push or thrust that must be produced by the engine if the propeller were 100% efficient. But the highest possible efficiency is 74%, seldom reached in practice, the best average being 67%. The average horsepower that must be delivered to the propeller shaft (termed *SHP*) is thus at least one-third larger than *EHP*. Owing to many losses, indicated horsepower *IHP* can only deliver 80% to the propeller, in which case the efficiency is $0.67 \times 0.80 = 0.535$ or 53.5%. This percent is termed the over-all efficiency of propulsion. Modern steam turbines and Diesel engines cannot be measured by *IHP*, but always by *SHP*.

From formula (9) it is clear that we must first estimate the resistance in order to find the *SHP* of the engine. The total resistance is made up by frictional and pressure resistance, as previously described. The former is dependent on the wetted surface *WS*, the latter on the angles of incidence at bow and stern, termed bow angle and stern angle, and on the area of the mid-section. *WS* = girth at mid-section × length ×

a constant; and from Fig. 7 we obtain girth = 94.10 ft., length = 326 ft., and the constant = 0.795. Therefore, $WS = 94.10 \times 326 \times 0.795 = 24{,}350$ sq. ft. (approx.). Frictional resistance $= C \times WS \times V^2 = 0.0070 \times 24{,}350 \times 13^2 = 28{,}900$ lbs. (approx.) at 13 knots.

Pressure resistance has been widely investigated by the late U. S. Admiral D. W. Taylor, but his values were unfortunately affected partly by error in their reduction from total R of the 20-ft. models used in his experiments, and partly by the effect of the basin walls on the resistance. Corrected, the pressure resistance per ton of displacement = 1.51, and $R_p = 1.51 \times 7{,}070 = 10{,}675$ lbs.

There are here two corrections to be made. The frictional resistance does not include that of bilge keels often fitted to sea-going ships to reduce rolling, and for which an addition of 3% is usually made, so $R_f = 28{,}900 + 870 = 29{,}770$ lbs. Further, the Taylor experiments were made in fresh water, and for salt water an addition of 2.5% should be made, so $R_p = 10{,}675 + 255 = 10{,}930$ lbs. Now add, and use formula (9),

Skin resistance including bilge keels	29,770 lbs.
Pressure resistance, salt water	10,930
Total resistance	40,700 lbs.

$$EHP = 40{,}700 \times 13 \times 0.00307 = 1{,}624$$

and $SHP = \dfrac{1{,}624}{0.67} = 2{,}425$

For North Atlantic service, add 15%; $SHP = 2{,}800$ (approx.).

Admiralty Constant. Needing a formula for quick estimate of horsepower, the English Admiralty developed a formula that includes only displacement and speed, and assumed that the resistance varied as the square of the speed. The formula is wholly incorrect except at very low speeds that are commercially unprofitable, and can be used only if derived from some known ship data, for small differences in displacement and speed. For shaft horsepower, the formula becomes

$$SHP = \frac{D^{2/3} \times V^3}{\text{constant}}, \text{ or constant} = \frac{D^{2/3} \times V^3}{SHP} \qquad (10)$$

$$\text{The constant for our design} = \frac{7{,}070^{2/3} \times 13^3}{2{,}800} = 290$$

SHIP'S WEIGHTS

In order to find the position of the engine room, it is necessary to estimate the weights and the centers of gravity of the different items that make up the displacement of the ship, 7,070 tons. The dead weight being 5,000 tons, there is left 2,070 tons for hull and machinery weights. Hence, if the weight of either one can be found, the other one is known too. The machinery weight is more easily estimated at so many lbs. per *SHP*, 196 to be exact. So, $196 \times 2{,}800 = 550{,}000$ lbs. $= 245$ tons (approx.). The weight of the complete hull $= 2070 - 245 = 1825$ tons. The several weights must balance on the center of buoyancy as a fulcrum. If the length of the afterbody was equal to that of the forebody, the center of buoyancy would be very nearly amidships, but in our design it has been moved forward because the afterbody is longer than the forebody. The former is 126 ft. and the middle body is 101.5 ft. By adding together $126 + 101.5/2 = 176.75$, and deducting one-half of $LWL = 163$ ft., the C.G. of the middle body is seen to be 13.75 ft. forward of amidships. Owing to the shape of the curve of areas, the center of buoyancy of the two ends of the vessel moves 36% of the C.G. of the middle body, 13.75 ft. Then, $0.36 \times 13.75 = 5.0$ ft., the distance forward of amidships of the center of buoyancy of the two ends. We know the displacement of all three divisions of the hull, and the rule to find their common center states:

"the moment of all parts divided by the displacement, equals the distance of the common center of buoyancy from the fulcrum."

In our case the fulcrum is amidships, and the computation shows the center of buoyancy to be 8.85 ft. forward of amidships. The student should make the computation observing that the lever of the ends is 5 ft., and the lever of the middle body is 13.75 ft., both from amidships as stated above.

To get the correct trim of the ship, the dead weight (5,000 tons) must balance the hull and machinery weight (2,070 tons) on the center of buoyancy as the fulcrum. The D.W. center of gravity is usually 1.75% of $LWL = 5.6$ ft. *forward* of the center of buoyancy, hence the C.G. of the remaining weights must be *aft* of the center of buoyancy. Using the rule just stated,

$$\text{C.G. of hull and machinery} = \frac{5000 \times 5.6}{2070} = 13.5 \text{ ft. aft of C.B.}$$

The C.G. of the complete hull with a cruiser stern is always amidships except where the machinery is placed aft, and if used as the new fulcrum, the hull moment = 0, and the C.G. of the machinery is easier to estimate. The center of buoyancy being 8.85 ft. forward of amidships, C.G. of hull and machinery thus becomes 13.5 − 8.85 = 4.65 ft. aft of the middle of *LWL*. The moment of hull and machinery = 2070 × 4.65 = 9625 ft.-tons, and if divided by the weight of the machinery 245 tons, the result = 39.3 ft., the distance of the C.G. of the machinery aft of *LWL* ÷ 2, that is, amidships.

The machinery consists of two Diesel engines of 1,400 *SHP* geared to the single propeller shaft by gearing, their weights being 106.5 tons and 20.5 tons. The shafting, propeller, and bearings amount to 9.4% of the machinery weight = 23.0 tons. The auxiliaries in the engine room weigh 0.034 tons per *SHP*, or 95.0 tons. This data can be obtained from the engine manufacturer, as well as the C.G. of the engine alone, which is at the middle of its length. Take the moments to this point, then *C.G.* of the machinery:—

Item	*Weight*	*Lever*	*Fore Moment*	*Aft Moment*
Two Motors	106.5	0		
One Gearing	20.5	15 ft.		522.0
Shafting, etc.	23.0	50		1150.0
Auxiliaries	95.0	6.7	640	
Totals	245.0		640	1672.0

The aft moment being the larger, clearly the C.G. is aft of the middle of the engine. Deduct fore moment from aft moment, or 1672 − 640 = 1032, divide this by the machinery weight 245 tons, result 4.22 ft., which is the distance of the C.G. of machinery aft of the middle of the motors. The length of the engine room is 42 ft. between the bulkheads with a 10-ft. recess aft for the gear casing. The engine plan shows that the middle of the motors is placed one-third of the length of the engine room, 14 ft., forward of the after engine room bulkhead. The position of the latter is 14.00 − 4.22 = 9.78 ft. aft of the C.G. of the machinery which is 39.40 ft. aft of *LWL* ÷ 2, hence, the after engine room bulkhead should be placed 39.40 + 9.78 = 49.18 ft. aft of amidships. The spacing of the frames in the ship is 25 ins., and the engine room extends over 20 spaces to within 7.18 ft. of *LWL* ÷ 2. Forward of the engine room the rules prescribe a collision bulkhead 5% of the length between perpendiculars, or 16.0 ft. aft of the stem at *LWL*, and another bulkhead not more than 20% of *LBP* aft of the stem, or 64 ft. No compartment should be more than 100 ft. long. Aft, there are the

after peak bulkhead, about 18 ft. from A.P., and an intermediate bulkhead 64 ft. from A.P. The six main bulkheads are thus located, and all should extend to the shelter deck. The length between perpendiculars is 320 ft., the after side of the rudder post placed 6 ft. from the end of the *LWL*. In this position there will be ample space for a 16-ft. propeller.

In connection with Fig. 8, the stem and the cruiser stern profiles are shown in Fig. 9.

CARGO SPACE. The total length of the cargo space is 244 ft., its width inside cargo battens 51 ft., its height 38 ft. amidships. With allowance for sheer and for camber of shelter deck, and deducting the shaft tunnel space, the cargo space occupies 80% of the block, or 280,000 cu. ft. The D.W. of 5000 tons includes fuel, water, and stores, amounting to 700 tons, leaving 4300 tons for cargo for which there is a space of 280,000 cu. ft. Each ton of cargo thus occupies 280,000 ÷ 4,300 = 65 cu. ft. Cotton in bales strongly compressed take up about 60 cu. ft. per ton, hence the ship can be filled and fully loaded with cotton; but if the cu. ft. per ton is over 65, the full cargo cannot bring the ship to its load line marks. Grain, on the other hand, takes up only 46 cu. ft. per ton, and the ship's cargo spaces cannot be filled by it, still less with iron ore at 9.2 cu. ft. per ton. A table of principal materials for bulk cargoes is added below.

TABLE III

WEIGHT OF BULK CARGOES IN CUBIC FEET PER TON

Materials	*Cu. Ft. per Ton (2240 lbs.)*	*Remarks*
Fresh water	35.9	Generally taken as 36, which is erroneous
Salt water	35.0	
Cotton bales	87.0	Light American cotton
Cotton bales	60.0	Strongly compressed
Coal	43.0	
Fuel oil	38.0	
Cargo oil	38.5	
Sugar in bags	41.5	
Rice in bags	50.0	Varies with size of bags
Wool, compressed	84.0	Washed and strongly compressed
Iron ore	12.0	Heavy Swedish ore
Grain, loose	46.0	Shrinks 8% during voyage, feeder spaces compulsory

> In all cases the *net* cargo spaces are used to figure the weight of the cargo, free from obstructions, and measured to the outside of the frames, and ⅔ up the beams for grain. For bags and bales, the space is measured to inside of cargo battens and to bottom of beams.

Water Ballast. In light condition without cargo, all ocean-going ships need ballast, for which a double bottom is built into the vessel. In addition, deep tanks are fitted in the hold that can be used as cargo space when needed, although not very useful because they are cut up by swash plates, deep frames, and girders. Deep tanks must always be kept filled by feeder trunks, to avoid swashing and straining the tank bulkheads; but double bottoms are seldom filled to capacity because, in cases of grounding, the inner bottom might be buckled and split open.

The amount of water ballast should be 25% of the cargo weight, or $0.25 \times 4300 = 1075$ tons if deep tanks are fitted, and 15% or 645 tons if without deep tanks, the latter weight being a minimum. In fast vessels that consume an enormous amount of fuel oil from the double bottoms, water is often pumped in as the oil is pumped out, but not without danger.

Details of Hull Weights. The hull weights are made up of steel, wood and outfit including deck machinery, cement and paint. The steel weight includes the longitudinal framing and plating, the transverse framing, castings and forgings, bulkheads, shaft alleys, engine seatings, bunkers, casings, and deck houses. Very little wood is now used in the living quarters, but wood decks are still laid, and wood is used for cargo battens, ceilings, and deck house tops. Equipment and outfit include cargo winches, anchor gear, spars, booms, and rigging, steering gear, ventilation, drainage, plumbing, fresh water systems, electric lights, furniture, life belts, sails and awnings, etc. Cement is applied in narrow spaces at the ends of the ship, to insure good drainage; and paint covers everything else.

Many formulas have been proposed for preliminary estimates of all weights but curves based on the principal dimensions are more accurate and quicker to use. Yet the Swedish Naval Architect J. Johnson developed formulas for weights of steel hull[4] that were widely used, although with stronger steel and new ideas in construction, the weights given by these formulas are too high. As a basis for comparison of

different ship types, the formulas are still useful, and are enumerated here.

WEIGHTS OF STEEL HULLS

by

J. Johnson, N.A.

Formula, $W = CN^x$

W = finished weights in tons of steel in hull
C = a constant from the Table below
N = Lloyds Old Longitudinal Numeral modified as follows. In three-decked vessels, the girths and the depths are measured to the upper deck without any deductions, in spar- and awning-decked vessels, to the spar and awning decks, in one-, two-, or well-decked vessels to main decks.
x = an exponent showing how the weights vary with the numerals.

Type of Ship	*Deductions*	*Formula*
Three decks with complete shelter deck	Upper tween-deck height	$W = 0.000359\ N^{1.48}$
Three decks	0	$W = 0.00078\ N^{1.40}$
Spar deck ships	0	$W = 0.00115\ N^{1.35}$
Awning deck ships	0	$W = 0.00167\ N^{1.30}$
One- , two- , and well-decks	0	$W = 0.00215\ N^{1.30}$
Sailing vessels	0	$W = 0.00065\ N^{1.40}$

Lloyds Old Longitudinal Numeral was obtained by adding the vessel's half breadth, half girth (from keel to gunwale) and depth (molded), and multiplying by the vessel's length from front of stem to back of sternpost.[5] In case of vessels exceeding a depth of 17 ft. and having three decks, 7 ft. was deducted before multiplying.

However, even curves do not always give correct results, as shown by Bartlett's.[6] Mr. Bartlett, a Surveyor at Lloyds, spent a tremendous amount of work in developing his curves based on the surface of a ship. But either the ship's dimensions were at fault, or the computations—in any case the estimated weight seldom agreed with actual weights.

The author has spent many years of research work on *steel weight curves* that should agree with *actual weights*, and the result is shown in Fig. 10 for shelter deck vessels, the most common type built at present. The curves are based on the principal dimensions: length B.P., beam, and draft, divided by 100 to avoid very high numerals that run from 1,500 to 24,000. On the curves are indicated the numerals for 3 to 6

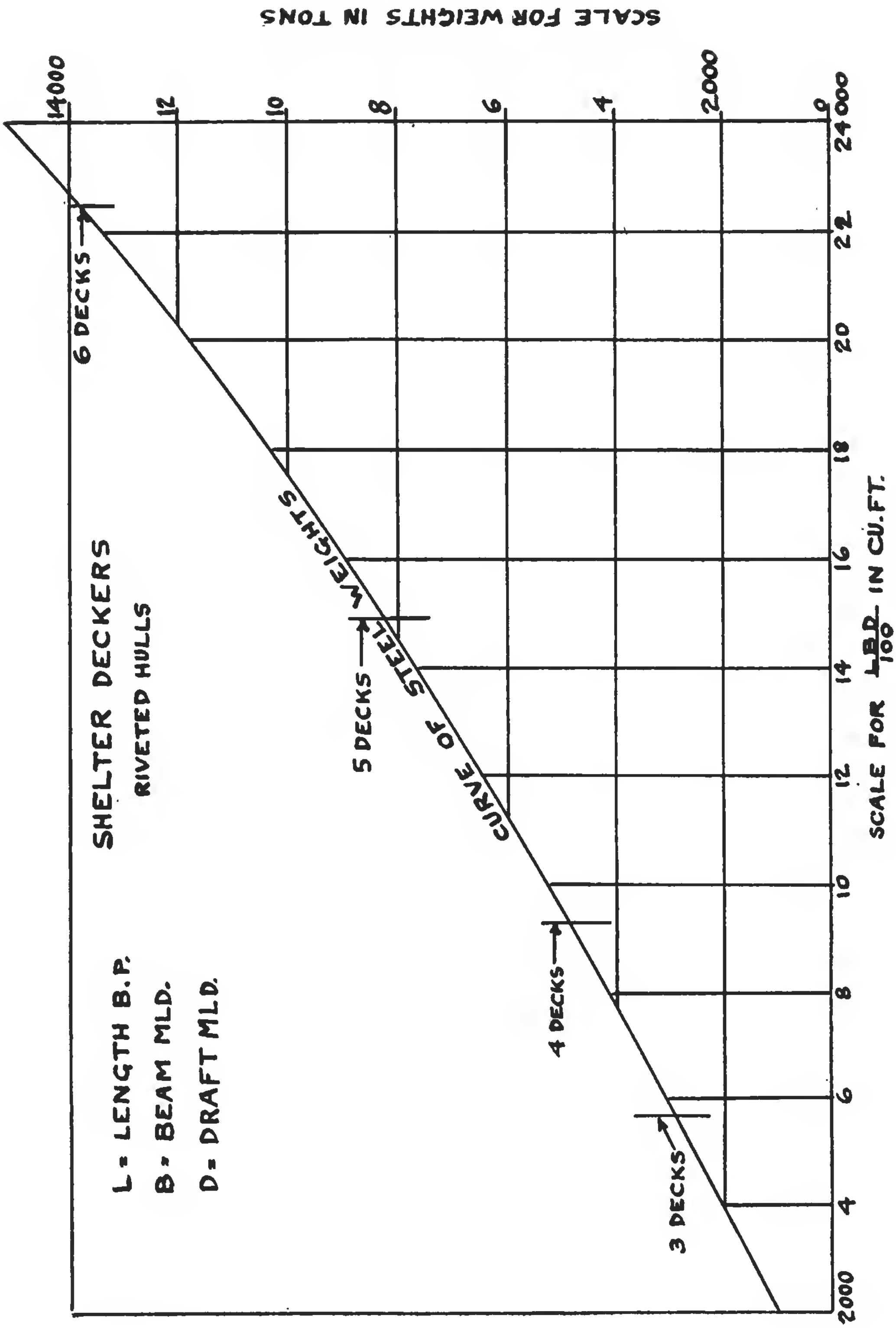


FIG. 10. STEEL WEIGHT CURVES.

decks besides inner bottom. In theory there should be a small step in the curve at each deck, but actually the curve is fair with very little curvature. Both curves are for riveted hulls; if welding is used, deduct 16% for all-welded vessels, and less in proportion for partial weldings.

For our design

$$\frac{LBD}{100} = \frac{320 \times 52.5 \times 21}{100} = 3525 \text{ (approx.)}$$

and the corresponding steel weight = 1680 tons. 16% of this = 270 tons (approx.), and net steel weight = 1410 tons. The total hull weight being 1825 tons, this leaves 415 tons for wood, equipment, and outfit, which is ample, even considering the extent of deck and other machinery now installed to save manpower. In passing, it might be added here that when steel was first used in shipbuilding, the general idea was that such ships must sink like a stone. The British post office declined for years to send mail by iron ships, and only 100 years ago at that.

The weights of equipment, wood, and outfit can be obtained from formulas based on the Equipment No. of the American Bureau of Shipping. It is made up of a volume $0.75 \times LBH \div 100$, plus a certain percentage of superstructures and deck houses that add about 13%, a minimum for cargo ships. For passenger ships, the addition must be figured from the plans, or assumed preliminary.

The Equipment No. of our design = $0.75 \times (320 \times 52.5 \times 30.83) \div 100 = 3890$ plus 13% = 505, total 4385 (approx.), and the formula,

$$\text{equipment, wood, and outfit} = C \times (\text{Equipment No.})^{2/3} \quad (11)$$

For cargo ships $C = 1.37$, and weight $= 1.37 \times 4385^{2/3} = 367$ tons. As there was left 415 tons, our estimate is on the safe side by 48 tons that may serve as a margin of over 4% on hull and machinery weight, 2070 tons. Other values of C in formula (11) are:

	C
Pacific liners carrying cargoes	3.15
Atlantic liners carrying no cargo	9.50
Auxiliary sailing ships, schooners	1.80
Square riggers, with mast winches	2.00
Ocean-going tugs	1.62
Harbor tugs	0.95
Great Lakes vessels and coasters	0.90
River towboats with comfortable quarters	1.55

The factor 0.75 in the Equipment No. is a maximum, and is reached only in very full vessels. The depth is really measured to the freeboard deck, and the shelter deck volume added, but the error is always on the safe side by including the shelter deck in the depth. Superstructures are treated separately in the American Bureau Rules, and added to the hull volume.

DISPLACEMENT–SPEED

In formula (7), page 32, were given the most favorable speed-length ratios from the shipowner's standpoint, and the ratio 0.72 specially commended. Within any speed-length ratio there is a certain displacement connected with every speed, for which combination the lowest possible resistance per ton of displacement is assured, and hence the lowest possible engine power and fuel consumption. Numerous attempts have been made to find this combination, generally based on the *Law of Similitude*, that never stood the test of practice. For similar ships, displacement grows as the cube of length (L^3), but with the ship proportions limited by stability, the displacement increases only as $L^{2.5}$, a big difference actually. Now L grows as V^2 and consequently, displacement increases as V^5, and not as V^6 according to the *Law of*

TABLE IV

SHIP PARTICULARS FOR SPEED-LENGTH RATIO OF 0.72

Sea speed	10	13	16	19
Length on LWL	193	326	495	696
Length between PP's	189.4	320	486	683
Beam at LWL	37.1	52.5	69.5	87.1
Draft molded	13.8	21.0	28.6	36.3
Depth molded to shelter deck	21.1	30.8	44.0	56.5
Prismatic coefficient	0.643	0.700	0.740	0.765
Mid-section coefficient	0.985	0.985	0.985	0.985
Displacement, tons	1785	7070	20550	47360
Dead weight	1238	5045	15015	33785
Shaft horsepower	533	2530	9330	30130
Revolutions per minute	114	104	94	107
Crew, two shifts	9	30	60	166

NOTE. The 19-knot ship is fitted with twin screws. All have Diesel machinery. Shelter deck is strength deck. Dimensions for other speeds interpolated. Table particulars produce ships of lowest engine power and highest carrying capacity per ton of displacement, as explained in Part III-B, page 179 *ff*.

Similitude. All deductions based on this law are thus entirely misleading, and although some textbook authors mention it, they have not stressed it enough, for to this day the *Froude Constant System* is used by some designers with disastrous results.

Displacement-speed is a new concept that deserves its place with other factors in the art of efficient ship designing. All ship particulars in Table IV are derived from the displacement-speed factor, once speed or length is assumed.

OTHER DESIGN DATA

DETAILS OF MACHINERY WEIGHTS. Estimates of machinery weights belong properly to the Engine Department, but the ship designer should be able to apportion accurate percentages for it in his ship weight tables or diagrams. Every piece of machinery consists of two classes of parts—rotative and fixed. The former includes all parts that move when the engine is running, the latter, all stationary parts. For a certain power, say one horsepower, it is evident that the faster a part moves or rotates, the lighter is the weight, for the same work can be done by a small engine at 1,000 revolutions per minute as by a big engine at 100 revolutions. The rotative parts of the small engine would weigh only 1/10 of those of the big engine if both were of the same type. But the fixed parts of an engine, such as bed plates and cylinders, are more independent of speed, and when the primary sources of power are considered, such as boilers or heaters, speed is without effect at all. It means that the weight per *SHP* of the boilers is the same for the small as for the big engine, but the total weight of the machinery is quite different in the two instances.

In his monumental work, *Design of American Superliners*,[7] Theodore E. Ferris made painstaking estimates of all weights from which can be deduced that the rotative weights amounted to 36.5% of the total machinery weights of 8175 tons and 146,000 *SHP* at 160 revolutions. The total weight = 125.5 lbs. per *SHP*, and the rotative weight = 45.8 lbs. which divided by the revolutions = 0.286 lbs. per *SHP/N*. These figures represent wet (including water) weights, and are probably the minimum obtainable in merchant ships. Compared to the weights of our design with 2,800 *SHP* at 100 revolutions,

$$\begin{aligned} \text{the total weight} &= 193 \text{ lbs. per } SHP \\ \text{the rotative weights} &= 71.5 \text{ lbs. per } SHP \\ &= 0.715 \text{ lbs. per } SHP/N \end{aligned}$$

considerably higher on account of the lower number of revolutions. All these values are for geared turbines with water-tube boilers and a steam pressure of 400 lbs. per sq. in. but, actually, Diesel engines would be 10% lighter in weight. However, the latter varies so much by type and make that actual data should be obtained from the manufacturers.

It was mentioned that the fixed part of the engine was more independent of revolutions than the rotative part. By combining the two, reducing the effect of speed, and by including the steam pressure, a more accurate formula can be developed. The weights of the boilers, etc., are proportional to the indicated horsepower, assumed to be 25% higher than the shaft horsepower *SHP*. The steam pressure has been found to affect the engine weights inversely as the cube root of the pressure in lbs. per sq. in. plus 15 (the atmospheric pressure at the seashore). By adding 20 to the revolutions per minute, the effect is lessened. The machinery weight of geared turbines and water-tube boilers, including shafting and auxiliaries,

$$\text{M.W.} = 279 \times \frac{SHP^{0.75}}{(N + 20)\sqrt[3]{P + 15}} + 0.231 \times IHP^{0.80} \qquad (12)$$

Assuming a steam pressure of 400 lbs., the cube root of 415 = 7.45, $N + 20 = 120$, $SHP = 2800$, $IHP = 3500$,

$$\text{M.W.} = 279 \times \frac{385}{120 \times 7.45} + 0.231 \times 685 = 278.6 \text{ tons.}$$

For Diesel engines deduct 10% = 250 tons, very close to our estimated weight of 245 tons as obtained from manufacturers. With higher steam pressures and revolutions, still lighter machinery could be produced, but high revolutions mean low propeller efficiency and higher *SHP* for a given resistance, and a balance between revolutions and efficiency must be struck as has been done here. Formula (12) has been developed from the fundamental *PLAN* for power, and $P_{max}LA$ for weight, where

P = mean pressure in the engine,
P_{max} = highest pressure,
L = stroke of engine,
A = area of piston,
N = number of revolutions per minute.

The exponent 0.75 of *SHP* makes allowance for the materials in the

engine, because castings of small engines must be almost as thick as in larger engines.

Revolutions. The screw propeller is given a greatly variable efficiency by changing the revolutions. There is a maximum at about 10%, too low for weight considerations except in the very fastest vessels. From this maximum, sometimes over 0.74, the efficiency is gradually reduced to zero by increasing the revolutions. There is a certain connection between speed, power, and revolutions that can be expressed in a formula once the efficiency has been determined upon. Accepting 0.67 as best for merchant ships, the formula becomes,

$$N = 19.7 \times \frac{V^{5/3}}{\sqrt[3]{SHP}} \tag{13}$$

or, revolutions = speed in knots raised to 5/3 or 1.67th power, divided by the cube root of SHP, and multiplied by a constant. For our design $N = 19.7 \times 71.5/14.1 = 100$ revolutions. The development of formula (13) was too long and too difficult to describe here.

Fuel Capacity. Diesel engines consume 0.39 lb. per SHP-hour, but in practice it is safer to make it 0.40 lb., and then add 15%. The daily consumption is $0.40 \times 24 = 9.60$ lb. per SHP, not including the 15%, and for 2800 SHP it amounts to 12.0 tons per day. For 30 days and 15% added, the weight of the fuel is 415 tons (approx.), and it occupies 17,220 cu. ft., to which must be added about 6% for obstructions inside the tanks, 18,250 cu. ft. in all, the tanks assumed to be filled to 97% of capacity. In addition to the double bottom tanks, lubricating oil tanks, deep tanks and settling tanks forward of the engine room need to be considered. The student should examine closely the modern engine rooms published in many marine journals.

Length of the Engine Room. Enough space must be had around the engine for working, and the length of the engine room is dependent on the engine length plus about 7 ft. As a first estimate, the beam of the ship can be taken with Diesel machinery, but only 80% needs to extend the full width of the ship, as explained on page 42. With steam turbines, less engine room length is required, about 75% of the vessel's beam, but little is gained, because the boiler fuel oil needs 50% more tankage than Diesel fuel oil, besides fresh water reserve feed

tanks. In both cases, high-speed vessels are given much longer engine rooms—for instance 2.5 times the beam in the Ferris' Superliner,[7] with a speed-length ratio of 0.95. Little space is left for cargo.

MANNING SCALE. For cargo ships, manning scale is usually based on D.W. tonnage, but a better way is to base the scale on gross tonnage. The crew is divided into three departments, deck, engine, and stewards, of which the first might be proportioned to the G.T., the second to the *SHP* of the engines, and the third is dependent on the other two in a cargo ship, and on the number of passengers in passenger vessels. Clearly one captain and one engineer are all the crew needed on the smallest vessel, and the two men are added separately to the number of the crew computed as follows:

Deck Department, 1 man per 360 Gross Tons, plus 1 man
Engine Department, 1 man per 330 Shaft Horsepower, plus 1 man
Steward Department, 1 man per 1700 Power Tonnage
(Power Tonnage = SHP + G.T.)

For passenger ships add 1 steward for every 6 passengers, and then add 25% of the total. When over 49 people aboard, add 1 junior radio operator. In case of 3-shift watches, increase deck and engine crew by 33%. For tankers deduct 10%.

AUXILIARY MACHINERY. Besides the main propulsion engines, the engine room contains machinery, mostly electric motors, for the electric lighting systems, for fuel transfer, for bilge and ballast pumps, for ventilation, etc., all in duplicate. Aft over the rudder is the steering engine, electro-hydraulic for larger ships over 3000 G.T., hand-operated in small vessels. On deck is a large variety of engines for cargo handling, anchor and mooring winches and capstans, lifeboat winches and davits, towing machinery, sounding and logging devices, etc. In passenger vessels, ventilating machinery is often located on the highest deck, and sometimes there is mechanical ventilation of the cargo holds (always in tankers) in cargo ships.

From the shipowner's standpoint, the cargo handling outfit is most important but, curiously enough, also the most neglected. A ship can only earn money at sea; every day spent in a harbor is a dead loss, although a necessary one. To shorten the turn-around period, the cargo ship is fitted with two or four king posts to every hatch, each king post

supporting two cargo booms served by 5-ton electric winches in small deck houses around the king posts. In this manner the turn-around period has been very much reduced when package goods are carried.

Ventilating and Heating. A cargo ship is liable to encounter all climates, and while ventilating in cold weather can be obtained by natural means if the living spaces are heated, in hot weather, fans and fan ventilating are necessary for the comfort of crew and passengers. In a Diesel ship hot water heating is supplied by a boiler using exhaust heat from the main engines at sea, and by oil firing the boiler when in harbor. There are now many systems of air conditioning for ships, as well as for houses on land, that should be studied by the ship designer. Both ventilating and heating require ducts and piping that greatly complicate the inside lay-out of a vessel.

LAUNCHING

Before a ship can be transferred to its element by launching, its stability under various conditions must be ascertained (see Part III). It has happened that a vessel with ample stability for sea-going nevertheless has capsized when launched. But as launching calculations are fairly simple, a description of the procedure is given here.

There are many things to be considered at launching—the ship, the cradle, the standing ways, the releasing and the checking or snubbing arrangements, besides stability and strength. The latter is now in the hands of the Register Societies and need not be explained here. The weight of the ship must first be estimated, including dunnage and the cradle, in order to find the sliding surface of the cradle. It is usual to allow a pressure of 1.70 tons (3810 lbs.) per square foot of the cradle surface, a little less in summer, a little more in winter, depending on the consistency of the grease or the material used to reduce the friction between the cradle and the standing ways. When the time for the launching draws near, the standing ways (if not permanent) are laid with an inclination seaward of 9/16 inch to the foot if the keel has been laid with the same declivity. The distance between the ways is about one-third of the vessel's beam when its bottom is flat amidships, otherwise the distance may be less. When the standing ways have been planed and greased, the lowest course of the cradle is also planed and greased, and the cradle built up to the shape of the hull. In vessels

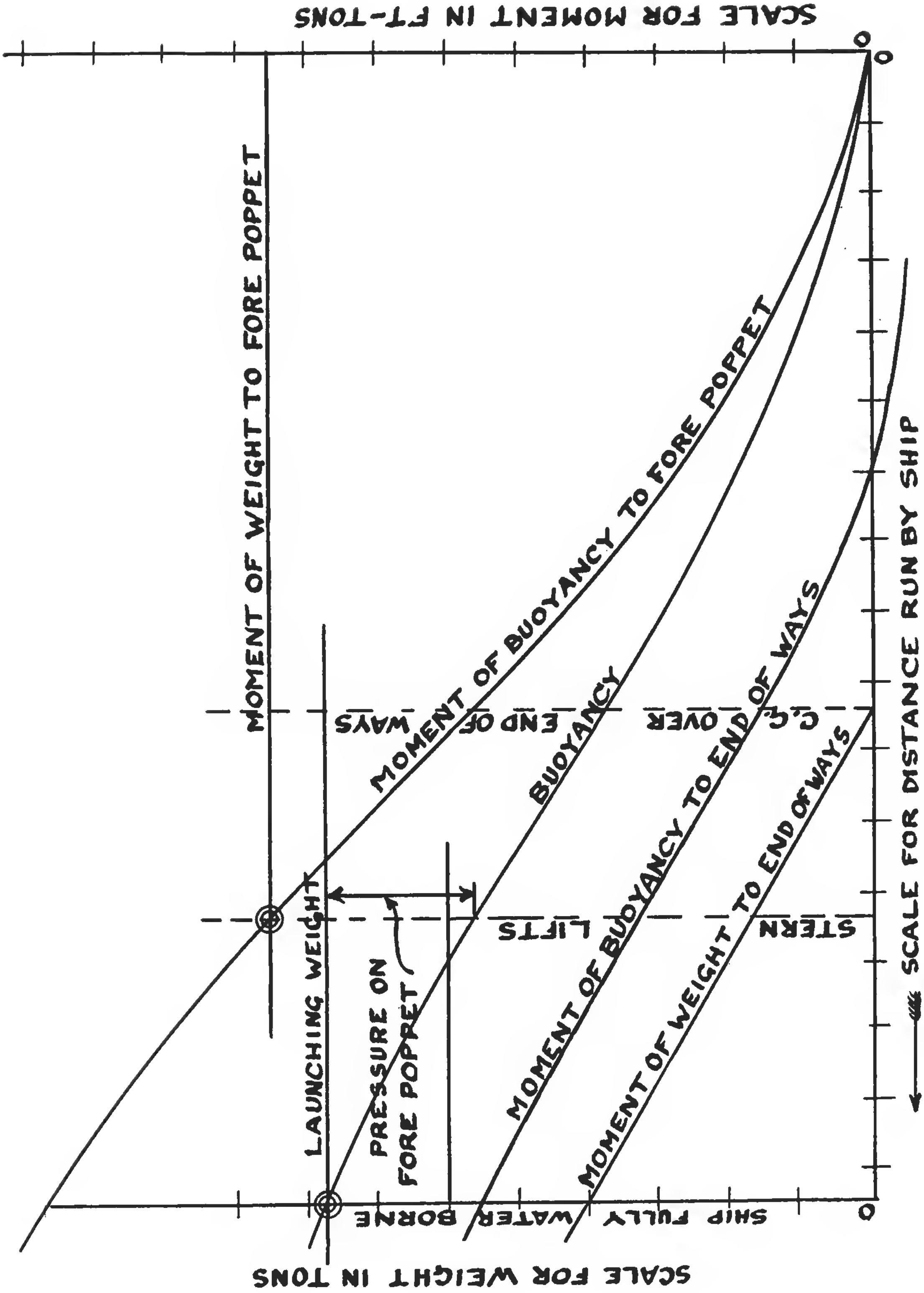


FIG. 11. LAUNCHING CURVES.

with fine ends it is sometimes necessary to connect the two parts of the cradle by steel plates under the keel, otherwise long steel cables are used, fastened by bolts to short uprights on the outside of the cradle. The standing ways and the cradle are always of wood.

As the ship starts to slide down the ways, there comes a moment when its after end is waterborne and begins to lift, increasing the pressure on the forward end of the cradle, or the *fore poppets*. Were it not for the resilience of the hull, this pressure would be tremendous, but actually it is seldom more than twice the average pressure. On the other hand, before the ship is waterborne, there is the danger of *tipping*, or pivoting about the end of the ways. This would happen if the vessel became waterborne *after* the C.G. of the ship and cradle has passed over the ends of the standing ways, and the result would be most disastrous. To guard against it, the ways are extended some distance aft of the position of C.G. when the after end is waterborne. The displacement of the after end and its center of buoyancy is readily calculated from the design; and its moment to the after end of the ways must always be larger than the moment of the ship's weight to the same point, in order to avoid tipping. If the curve of areas for different displacements of the after end is drawn, and its moment to the end of the ways is calculated (paper patterns are useful), then a curve of the moments for any position of the ship on the ways can be drawn. Similarly, a curve of the moment of the ship weight is also made, and the two compared. If the two curves should cross, there is a positive danger of tipping.

To ascertain the pressure on the fore poppet, the curve of areas is drawn for still greater immersions of the ship's after end, and its moment to the fore poppet for each immersion computed. Similarly, the moment of the ship's weight to the same point—this, of course, being constant. Immersion and position of the ship are coordinated, the declivity of the ways being known. As before, curves of the two moments are drawn, and where they cross the stern should begin to lift; but, owing to the high velocity and the momentum of the ship, the latter travels some distance farther before the pressure on the fore poppet becomes excessive. In the region of the fore poppet, a large ship is always strengthened by internal shoring or other means to distribute the pressure over the hull surface. Generally, the pressure is assumed to be equally distributed over the length of the sliding ways remaining when the stern lifts—clearly erroneous, since at this point

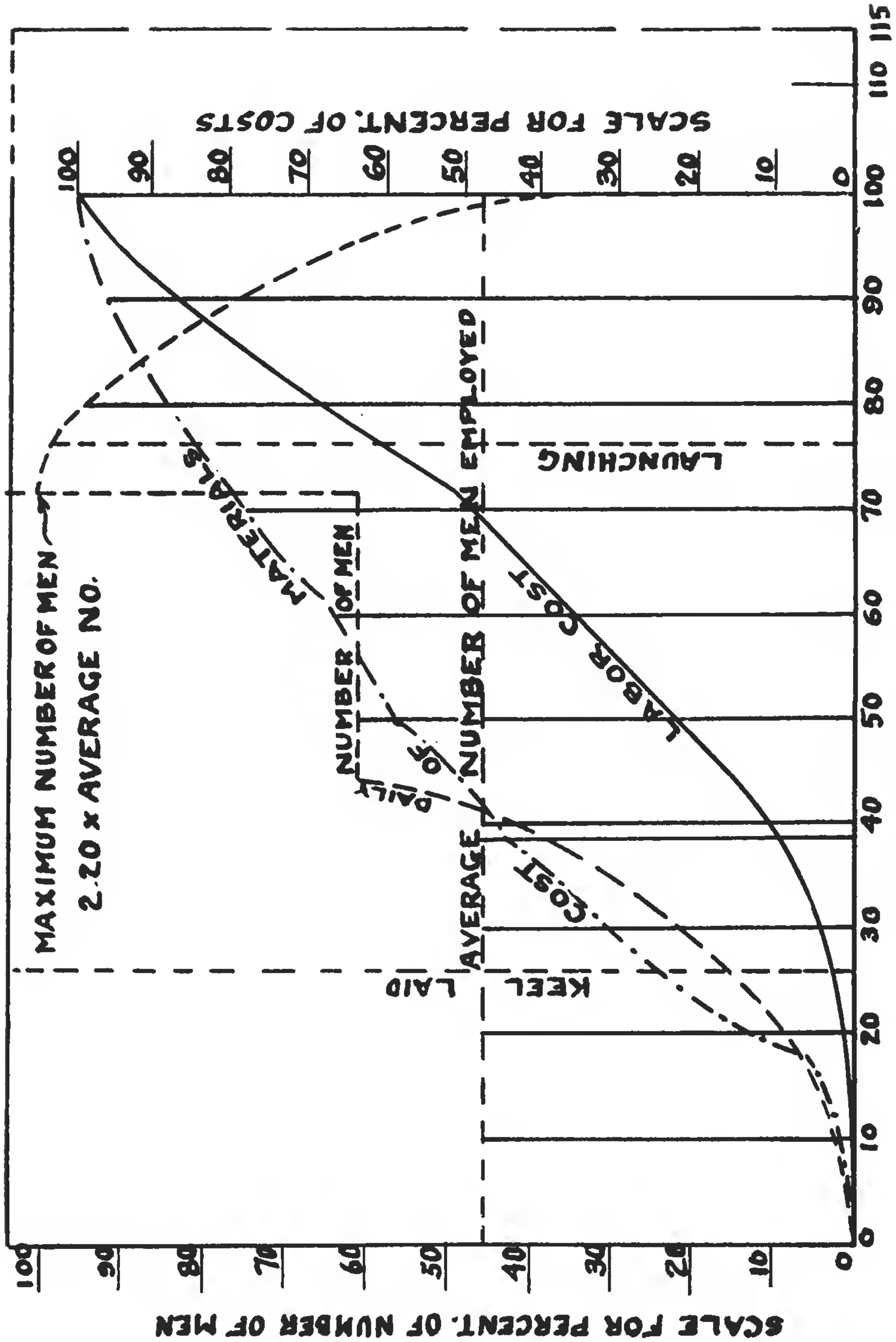


FIG. 12. PROGRESS CHART.

the pressure at the end of the ways must be *nil*. In this case, the pressure at the fore end of the cradle amounts to twice the average or still more but, owing to the resilience of the hull, we have no means to assess its actual value. The total pressure acting on the fore poppet is the difference between the ship's weight and the displacement or buoyancy at the point where the stern lifts, and it is best obtained graphically as in Fig. 11. On the base are marked off, to scale, the feet run or traveled by the ship on the ways. Vertically on the left is a scale of weight, and on the right a scale of moments, the former in tons, the latter in foot-tons, and the weights, the buoyancies, and the moments marked off at the different positions of the ship as already explained. Where the moment curves cross, the stern lifts, at which point the difference between the weight and the buoyancy (displacement) gives the weight on the fore poppet to the scale on the left. Note that there are two poppets.

If the space outside the ways is restricted, chain drags or anchor drags are used to stop the ship after clearing the ways, or side launching is resorted to.

In side launching, a great many ways are laid under the ship at intervals of 10 to 15 feet,[5] and several stoppers or dogshores are fitted, all dropped at the same time. The main object is to get the vessel clear of the ways by launching with considerable velocity. This is attained by a great declivity of the ways, up to 4 inches to the foot or more. Naturally, there is no need of intricate tipping or lifting computations.

PROGRESS CHART

During the building of a ship it is always necessary to know how many men to employ, and how much material to buy, in order to make delivery at the contracted time. For a working week of 5½ days, and allowing 26 days per year for absence, the net working week is made up of five working days as an average. To estimate the average number of men, divide the total number of working days by five times the number of working weeks from date of contract to delivery time. To estimate the total number of working days, divide the total labor cost by the average earnings per man per day. The total labor cost varies widely, between 25% and 45% of the contract price, but 35% can be taken as a good average.

For instance, the total working days have been estimated as 700,000, number of weeks 35, multiplied by 5 equals 175, hence average number of men, $\frac{700{,}000}{175} = 4{,}000$, and the maximum number $2.20 \times 4{,}000 = 8{,}800$ to be employed shortly before the launching of the ship. The dotted line in Fig. 12 represents the number of men employed daily during the building period. The number rises quickly at first, then it remains stationary for a considerable time, rises abruptly about a month before the launching time, and finally drops slowly to the time of delivery. Sometimes (always in warship building) there is some finishing work to be done after delivery that must be taken into account. The full-drawn curve shows how the labor cost, in percentage, is expended day by day until it rises to 100% at delivery time. In the labor cost, drafting is included but not shipyard officers' salaries. The uneven dash-and-dot line represents money spent for materials, and indicates that the buying is always ahead of the labor requirements, as it should be. This line is also on a percentage basis.

At the launching time a number of extra men are engaged, one man per 19 tons of the launching weight of the ship, or one man for each 2 feet of the length of the cradle. Their principal work is to wedge up the ship off the keel blocks.

In the case of an all-welded ship, there is a saving of 6% in labor cost, and 16% in the steel hull weight, thus reducing the time of delivery by about 10% but this depends on the number of men employed. In the German journal *Schiffbau* for 1930 there are some interesting progress curves for materials put into their latest big liners; but as the cradles are included in the weights, the curves are of no practical value for estimating steel requirements.

However, by taking the weekly steel rate per man, some estimate can be obtained of the efficiencies of the men and the shipyard tools. In American yards, the weekly steel rate amounts to 0.050 tons = 112 lbs. per man.

For our design, the estimated cost is \$250 per dead weight ton, \$1,250,000 in all. The labor cost at 35% = \$437,500 less 6% for the welded ship, \$411,250 net. Assume delivery in six months (26 weeks), then, if \$5.50 = average daily earnings

$$\text{Number of working days } \frac{411{,}250}{5.50} = 74{,}800$$

Average number of men $\dfrac{74{,}800}{5 \times 26} = 575$

Maximum number of men $2.20 \times 575 = 1265$

Weekly steel rate $\dfrac{1412}{26 \times 575}$ 0.0945 tons = 211 lbs.

TONNAGE ESTIMATE

The measurements for a ship's tonnage are usually made by some government department, such as the Custom House, but the ship designer should be able to make a close estimate of both the gross and the net tonnage. The *gross tonnage* is the capacity within the frames or ceiling of the hull of a vessel and of the closed-in spaces above deck available for cargo, stores, passengers or crew (with certain exemptions) expressed in tons of 100 cubic feet. The *net* or *register tonnage* is the remainder after deducting from the gross tonnage the spaces occupied by the propelling machinery (including allowances for fuel), windlass spaces, crew quarters, master's cabin, store rooms and navigation spaces, as defined by U. S. Department of Commerce. *No space can be deducted*, however, that has not first been included in the gross tonnage.

The estimate of tonnage begins with the determination of the tonnage deck which is the upper continuous deck in one- or two-deck vessels, and the second deck from below in vessels having three or more decks. The length on the tonnage deck is divided into 12 equal parts if the vessel is over 225 feet in length, and the area of each section within the frames or cargo battens, above the inner bottom, and below the deck, is computed and the capacity under deck obtained by Simpson's Rule. In the design state, ten sections should be enough for any length. The length is measured from the inside of the frames (or ceiling) at the stem to the inside of the same point at the after end of the deck. The position of the sternpost is not considered.

The capacity, so found, divided by 100, is termed the *under-deck tonnage* of the ship. If there are one or more decks above the tonnage deck, the 'tween-deck volume of each is estimated and together with the volumes of superstructures including deck houses, is added to the under-deck tonnage after dividing by 100. This constitutes the gross tonnage, but from it are excluded all spaces *above* the shelter deck used

for the handling of the ship, such as light and air ducts, staircases, toilet rooms, galley, pantry, etc., as well as spaces occupied by auxiliary machinery.

From the gross tonnage the necessary deductions are made as noted above, in order to arrive at the net tonnage, upon which all dues are paid. The deduction for the machinery space is the main one and a very peculiar one. If its tonnage is between 13% and 20% of the G.T. (gross tonnage), the engine room allowance is 32% of the G.T. whatever its actual volume. If under 13%, the allowance is only 1.75 times the actual volume. The shaft alley, the light and air spaces of the engine and boiler rooms are included in the machinery space, but not the fuel tanks or the bunkers. Such light and air spaces above the shelter deck can be included if strongly built and watertight. It is evident that the 13% proportion is the best from the shipowner's standpoint, but difficult to attain in a Diesel ship.

One other peculiarity of the tonnage rules* is the fact that the shelter deck space used for cargo, is not included in the G.T. if the shelter deck is fitted with tonnage openings either forward or aft in conjunction with scuppers and freeing ports near the deck below. However, if cargo is carried on any trip, the space so occupied is added to the tonnage of the ship.

The original idea of the shelter deck was to procure a shelter for cattle built on very light superstructures. Recently, the shelter deck is constructed as the strength deck, with the heaviest shell plating joined to it; and, as cargo is mostly carried in the 'tween deck, little is left of the original idea except the tonnage openings. But the improvement results in a lighter and stronger ship, and in the possibility of a lower net tonnage when heavy cargo is shipped.

STEERING

In land vehicles the steering is always done at the front, while in ships it is behind, although sometimes bow rudders are fitted to increase the maneuvering ability. The main reason for this difference in the steering methods is to be found in the turning axis position, near the center of gravity in the car, but near the stem in the ship when going ahead, and near the stern when backing. As the effect of the

* For the Suez Canal and the Panama Canal, there are special tonnage measurement rules that need not be considered here.

rudder is dependent on its distance from the turning axis, it can readily be seen why a ship steers so badly when backing.

The effect of the rudder comes from the water striking against its surface, causing an excess pressure on the front, and a suction on the back, the latter being the strongest. Along the edge of the rudder, there is a neutral zone where pressure and suction balance, for which reason small rudders are less efficient than big ones. The pressure on the rudder depends on its area, its angle of incidence with the stream flow, and mostly on the speed of the ship. The rudder area is usually 1.7% of the lateral plane in large vessels ($L \times D$) but increases to 2.5% and even 3% in tugs, river craft, and ferries. The maximum angle has been found to be 35° which curiously enough corresponds to the stalling or *burbling* angle of airplanes. There seems to be a decided change in the stream flow at this point, the water rushing in behind the rudder to reduce the suction. The speed of the flow past the rudder is affected mostly by the propeller race, and to a lesser degree by the wake and the stern wave motion. It is known that the water particles have a forward motion in the wave crest, and an aft motion in the hollow. For accurate estimates, these changes in the speed must be taken into consideration.

Many experiments have been made on rudders to connect the three factors, area, angle of incidence, and speed by a simple formula, the result being,[8]

$$\text{Rudder Pressure} = 3.2\, AV^2 \times \sin a \tag{14}$$

A = the rudder area in square feet

V = the speed of the flow in knots

Sin a = the sine of the angle of incidence between the rudder and the middle or symmetry plane of the ship

The coefficient 3.2 should be slightly smaller for small vessels, slightly larger for big ones.

The contour of the rudder is mostly rectangular with the aft corners rounded to cut away less effective parts. While the pressure is little affected by the contour, the twisting moment on the rudder head is dependent on its width, contour, and area. The center of pressure is not at the center of gravity of the rudder plate, but always ahead of it when the ship is going forward, and aft of it when backing. In the latter case, the twisting strains on the rudder head are generally maximum, the only exception being turbine steamers that cannot back very fast. For the greatest rudder angle of 35°, the center of pressure is at

46% of the mean width of the blade from the center line of the rudder stock when going ahead. Twin screw vessels whose propeller race only slightly affects the rudder, should be given about 10% larger rudders than stated above.[8]

The action of the rudder on the ship is not dependent on the total pressure of the rudder but only on its transverse component $P \times \cos a$, and on its distance from the turning axis. The cosine for 35° angle is 0.819, hence only 82% of the pressure is active in altering the ship's course. The path of the ship is at first a spiral, that later becomes a circle if the rudder is kept at the same angle. The diameter of the circle is termed the tactical diameter of the ship, and is important for maneuvering purposes, particularly in warships.

The dimensions of the rudder head, the pintles, and the tiller are now tabulated by the Registry Societies and need not be considered here.

Besides the usual rudder affixed to the rudder post of the ship, there is the balanced rudder in which part of the rudder area is forward of the center line of the rudder head, thus counterbalancing the after part and making it easier to turn the rudder. Sometimes a skeg or fin is attached to the rudder, with the balancing part wholly below the skeg, the latter being fitted with a bearing for the rudder at the lower end, and with pintles. When there are neither skegs nor bearings, the rudder is termed spade rudder, a very dangerous contrivance, even for a small boat, as it is liable to break and drop out. Besides, its skin resistance per square foot is larger than that of an ordinary rudder affixed to a sternpost, because it sticks out into water not acted upon by the frictional layer next to the skin of the hull. This fact is explained in Part IIIA, page 98 *ff*.

Finally, in corroboration of Formula (14), should be mentioned the extensive experiments made at the British model tank in 1932,[9] from which was deduced that the factors V^2 and sin a best agreed with actual results. The only difference was in the coefficient 3.2, which came out somewhat smaller, or only 2.33.

STABILITY

In order to estimate the stability of a vessel at any angle of heel, the three centers B_1, G, and M (Fig. 3, page 14), must be located in the body plan. At very small angles, G and M remain stationary, but B

travels to B_1, from which point a vertical plumb line meets M. At large angles, only G remains fixed.

The position of the center of buoyancy B can be obtained by estimating the areas of several water lines equally spaced between the *LWL* and the keel, and treating the areas as sections in Simpson's Rule—a very tedious procedure. A better method is to make use of the *curve of displacement* the area of which in square inches, divided by the displacement, gives the position of B below any water line. The curve of displacement indicates the displacement at any draft, and is drawn by offsetting horizontally the displacement in tons of 35 cu. ft. at several water lines or mean drafts. Such a diagram is always made for displacement calculations, and entails no extra work for the stability computations.

For example, if the displacement curve of our design is drawn to a scale of 1 in. = 800 tons, and a draft scale of 1 in. = 4 ft., and its area = 22.2 sq. ins., then at *LWL* the displacement is represented by a horizontal line 7070/800 = 8.83 ins., and the area divided by displacement = 22.2/8.83 = 2.51, and multiplied by the draft scale, 4 = 10.04 ft. as the depth of B below *LWL*. By taking the displacement at any other water line, the corresponding B is found in a similar manner.

There are several formulas for preliminary estimates of the position of the center of buoyancy, such as Normand's,[10]

$$B \text{ below } LWL = \frac{D}{3}\left(0.50 + \frac{C_b}{C_o}\right) \tag{15}$$

where C_b is the block coefficient and C_o the *LWL* coefficient. For our design, this works out as

$$B \text{ below } LWL = \frac{21}{3}\left(0.50 + \frac{0.688}{0.765}\right) = 9.80 \text{ ft.}$$

Having settled B, our next step is to find the position of M, the metacenter. In Part I, page 15, it was explained that to find M one must find B_1, the inclined position of the center of buoyancy. The shift from B to B_1 is caused by the wedge moment, that must be equal to the displacement moment $W \times BB_1$ in Fig. 3. The wedge moment is made up of the volume of either wedge (both are equal) multiplied by the distance between their center of gravity. For a very small inclination the wedge sections are triangular, their areas = $\frac{1}{2}\,bh$ = $\frac{1}{2}\,b \times$

$b \times \sin a = \frac{1}{2} b^2 \times \sin a$, where b is the half-breadth of the section. The distance between the wedge centers $= 2 \times \frac{2}{3} b$, and the moment of the wedges $= \frac{2}{3} b^3 \times \sin a$ per foot of the ship length and $\frac{2}{3} b^3 \times L \times \sin a \times$ constant for the whole length. The constant is derived from the summation of the different foot-pieces, and = 1.0 for a rectangular water line. Since a ship's water line usually is pointed at both ends, the ship's constant is always less than 1.0. It is connected with the coefficient of the water line C_o in a manner as shown in Fig. 13.

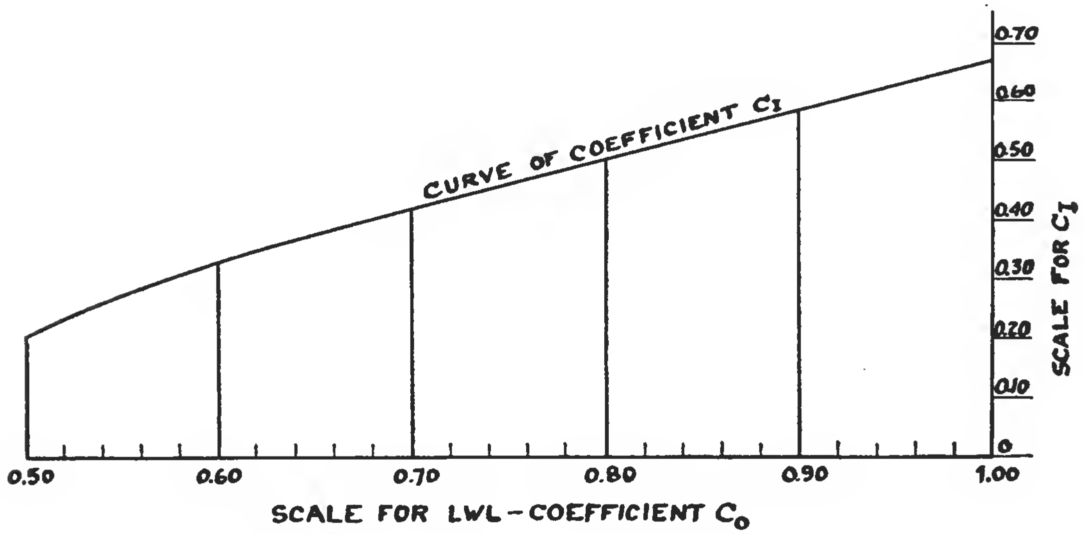


FIG. 13. COEFFICIENTS OF INERTIA MOMENTS IN RELATION TO COEFFICIENTS OF AREAS.

The product $\frac{2}{3} b^3 \times L$ is termed the moment of inertia for the water line in question. It is estimated by treating b^3 as ordinates in the Simpson's Rule.

The shift BB_1 is equal to the wedge moment divided by the displacement, $\dfrac{\frac{2}{3} \times L \times b^3 \times \sin a \times C}{W}$; MB the metacentric height, is very nearly $= \dfrac{BB_1}{\sin a}$. It is exactly $= \dfrac{BB_1 \times \cos \frac{1}{2} a}{\sin a}$, but since the angle a is very small, $\cos \frac{1}{2} a = 1.0$ practically, and the first equation for BB_1 is thus correct. Hence $\sin a$ is eliminated in the expression for BB_1, and $MB = \dfrac{\text{moment of inertia of } LWL}{\text{displacement in cu. ft.}}$

$$MB = \frac{C_I \times L \times B^3}{W} \tag{16}$$

There remains now only the position of G. It is possible to estimate the position of G by ascertaining the weight and the moment of every part of the ship, but it is very slow and tedious work. For preliminary purposes the distance from the top of the keel to G can be taken as a percentage of the molded depth to the uppermost continuous deck according to the following:

TABLE OF CENTER OF GRAVITY ABOVE KEEL IN PERCENT OF MOLDED DEPTH TO UPPERMOST CONTINUOUS DECK, SHIP FULLY LOADED

Type	*C_x of Mid-Section*	*Percent*	*Remarks*
Atlantic liners	0.95	0.63	Ferris' liner 0.60
Cargo, passenger	0.96	0.59	
Cargo, general	0.96	0.58	Poop, bridge, forecastle
Cargo, general	0.98	0.55	Shelter-decker with main deck = strength deck
Cargo, general	0.98	0.60	Shelter deck = strength deck
Tankers	0.97	0.54	
Small cargo	0.96	0.70	Poop, bridge, forecastle

Note. For small vessels a more exact result is obtained if one-third of the superstructure height is added to the molded depth after reducing for the length covered by the superstructure.

Returning to Fig. 13, the coefficient C_I (I is the symbol for inertia) refers only to the curved outlines of the water lines. If there should be any parallel middle body in the vessel, its moment of inertia is estimated separately and added to that of the ends. Its C_I is always 0.0833 when the full beam B is inserted in formula (16) as shown in Fig. 13, $C_o = 1.00$ clearly.

The metacenter estimated as here developed is termed the transverse metacenter. Its position influences the rolling of the ship, but there is also a longitudinal metacenter that greatly affects the pitching motion. Its position is calculated in the same way, but its C_I is only 88% of the transverse C_I in Fig. 13, for parabolic water lines. For elliptical *WL*'s there is no difference in C_I, HENCE, A FULL *LWL* IN THE STERN ADDS GREATLY TO PITCHING.

A very simple rule for the preliminary estimate of beam and draft

in proportion to length is the following,

$$\frac{B^2}{D} = C \times L^{2/3} \tag{17}$$

The constant C varies slightly with the height of the superstructure and the size of the ship. $C = 2.80$ is a good average for shelter deckers. Extreme examples are *Turbinia* with no superstructure, $C = 1.26$, and *Queen Mary* with layers upon layers of superstructures and deck houses, $C = 3.56$.

Example. The metacentric height of our design is arrived at as follows. $C.G.$ from top of keel $= 0.60 \times 30.83 = 18.50$ ft. B from keel $21.0 - 10.0 = 11.0$ ft.

Item	*Forebody*	*Afterbody*
C_o	0.625	0.695
C_I (Fig. 13)	0.0350	0.0415
Length	98.5	126.0

For the middlebody, $C_o = 1.0$, $C_I = 0.0833$, length $= 101.5$ ft. The beam $B = 52.5$ ft., the length $L = 326$ ft., and

Moment of inertia of water plane, forebody

$$= 0.035 \times \frac{98.5}{326} \times 52.5^3 = 15{,}330 \text{ (approx.)}$$

$$\begin{aligned} \text{middlebody} &= 37{,}600 \\ \text{afterbody} &= \underline{23{,}250} \\ \text{Total} &= 76{,}180 \text{ ft.}^4 \end{aligned}$$

$$\begin{aligned} M \text{ above } B \quad &= \frac{76{,}180}{7{,}070} = 10.76 \text{ ft.} \\ \text{center of buoyancy above keel} &= \underline{11.00} \\ & 21.76 \\ \text{center of gravity above keel} &= \underline{18.50} \\ \text{metacentric height} &= 3.26 \text{ ft.} \end{aligned}$$

Inclining Experiments. The position of the center of gravity is very difficult to estimate, yet of highest importance to the safety of the ship. For these reasons, every American vessel of 500 gross tons and over, carrying 50 or more passengers, must undergo an inclining experiment under government supervision. Its purpose is to ascertain the

metacentric height MG by shifting weights from one side of the ship to the other. The resulting angle of inclination is measured by the deflection of one or more plumb lines fixed in the centerplane of the ship. The tangent of the angle of inclination equals the deflection divided by the length of the plumb line. For instance, if the deflection is 3 ins. or 0.25 ft., and the plumb line 15 ft. long, tan a = 0.0167, and $a = 1°$ nearly.

W = displacement in tons, w = weights shifted, in tons, l = total distance of weights shifted, in feet—then, it is evident that the moments of weights must be equal, or $W \times GZ = w \times l$ (*see* Fig. 3),

$GZ = \frac{w \times l}{W} \times GZ$ also $= MG \times \tan a$, hence

$$MG = \frac{w \times l}{W \times \tan a} \tag{18}$$

Stability Diagrams. A cargo ship floats at a great many different drafts, and it is necessary to know the stability condition at each. It is, of course, not necessary to go below the light draft of the ship, in which condition the position of G is about 4% *higher* than in the loaded condition, except in tankers where G is only moved upwards 2%. The positions of B and M thus obtained, curves are drawn through each that show their position at any draft. The base is the molded draft in feet to scale, and the positions of B and M marked off to the same scale.

There are two other diagrams that depict the results of stability calculations. One shows the righting lever GZ for a constant displacement at every angle of heel up to 90°, the other shows GZ for varying displacement at constant angle of heel. The latter curves are termed *cross curves of stability*, and from them can be derived the former curve of *statical stability* at any displacement, but the righting levers must be corrected for any change in the position of G.

Besides these two curves, there is a third one termed *curve of dynamical stability*, that depicts the work in foot-tons expended when careening a vessel to any angle. As this curve concerns only sailing ships, it is mentioned here simply for completeness.

Trim. The difference in draft forward and aft of a ship is called trim. A vessel is on even keel when the draft is the same at both ends. It is

trimmed by the stern when the draft aft is the larger, otherwise it is trimmed by the head. The latter is a dangerous condition, as it makes the ship steer badly. Trim is caused by shifting or adding weights to the vessel, and is estimated by the effect of the weight moment on the position of C.G. after the weight is shifted, say a distance d. According to the Law of the Moments, $w \times d = D \times GZ_1$ where w is the weight

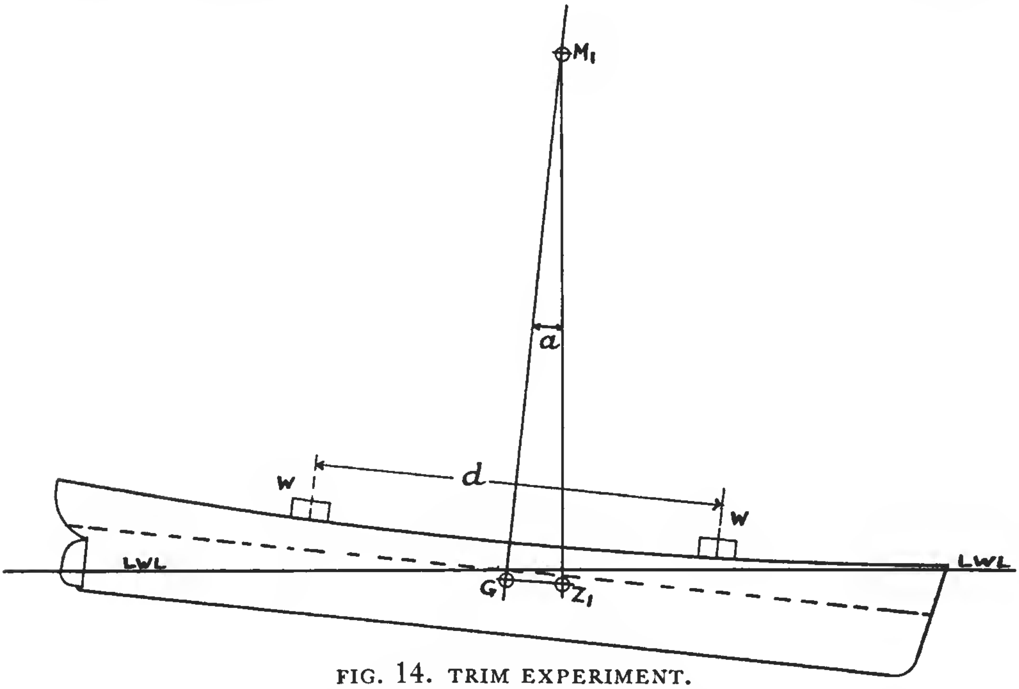


FIG. 14. TRIM EXPERIMENT.

shifted, D = displacement in tons, and GZ_1 the travel of G. As in the inclining experiment, $GZ_1 = M_1G \times \tan a$, (Fig. 14), and

$$w \times d = D \times M_1G \times \tan a; \tan a = \frac{w \times d}{D \times M_1G}$$

At small alterations of trim, that is, at small angles a, the ship swings longitudinally on an axis passing transversely through the center of gravity of the *LWL* plane, which is generally aft of the middle of *LWL*. The trim alteration at the stem is thus greater than that at the stern, in proportion as the axis is distant from the stem and the stern, or rather from the draft marks at these places. Call the distances d_1 and d_2, then $d_1 + d_2 = L$, and the total trim alteration $= L \times \tan a$. Insert the value of tan a obtained above,

$$\text{total trim alteration} = L \times \frac{w \times d}{D \times M_1G} \tag{19}$$

The weight moment to alter trim one foot is termed the *trim moment*, and is easily obtained from formula (19), and estimated for several different drafts of the ship. It equals $w \times d = D \times M_1G/L$. If the weights are added as well as shifted, or if taken out of the ship, the new displacement in tons must be used in the formulas, and the new M_1G, instead of the old values.

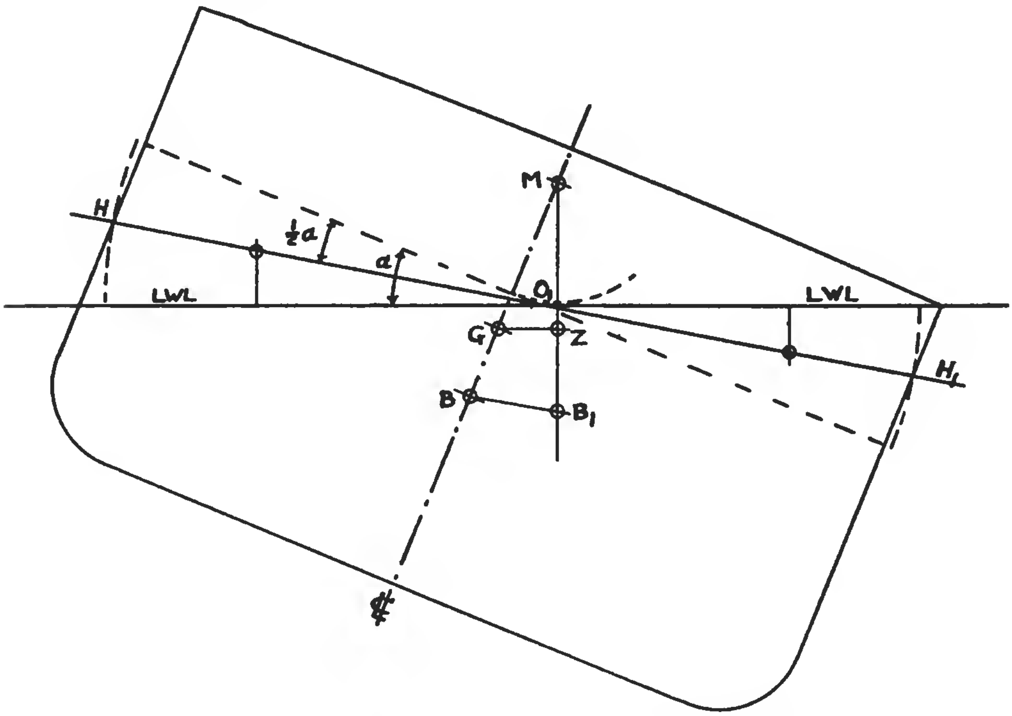


FIG. 15. STABILITY AT GREAT ANGLES OF INCLINATION.

STABILITY AT GREAT ANGLES OF HEEL. The formulas for stability estimates hitherto developed no longer hold at great angles of heel, principally because the metacenter M is lowered as the inclination becomes greater, until finally M falls plumb through G, ship's center of gravity, and the vessel capsizes. In the upright position, M falls plumb over G if the vessel is stable, and there are only these two positions to be reckoned with. To find the angle at which the ship would capsize, the *curve of stability* must be drawn with the angles of inclination as base, and the different righting levers GZ as ordinates. As G remains stationary, we have only to find B_1, the new position of the center of buoyancy in the *body plan*, and to extend a line vertically from B_1 (*see* Fig. 15).

There are many methods of locating B_1, most of them very time-wasting, hence not to be considered here. The simplest method is to cut out paper patterns for each full section in the body plan, and to glue the patterns together using the *LWL*, the center line, and the inclined water line as guides. Very little glue is used, and spread around the C.G. of each section. The C.G. is found quickest by balancing the pattern on a pin. When dry, the whole is hung up with plumb lines from two or more different points. Where the plumb lines cross is the position of B_1, to be carefully transferred to the body plan. By drawing a line square to the inclined water line, the new M and GZ are directly found from the body plan.

The inclined water line always crosses the upright *LWL* on the inclined side of the body plan—the farther out it is, the greater the angle. To locate the inclined *WL* with precision, first draw a short circular arc from M, touching *LWL* in the center line of the ship. The arc can be considered as a small part of the *curve of flotation*, that can be defined as the curve which every inclined water line must touch in order that the displacements shall be constant throughout. It is usually a fair curve up to the point where the inclined water line touches the deck of the ship, or where the deck becomes immersed. At this point, the curve of flotation is broken—has a *cusp*—and the ship sinks deeper in the water than indicated by the circular arc from M. At a 90° inclination a very close guess of the correct *WL* can be made by drawing a line in the body plan parallel to the center line, and at a distance from the latter of 10% of the freeboard F. Knowing the angle at which the deck becomes immersed, the cusp and its *WL* can be located by dropping a line from M at that angle. Intermediate *WL*'s can then be drawn in by the eye.

Of course, these considerations are not mathematically correct, but serve only as a close approach to the actual inclined *WL* at each angle used, and the actual displacement must always be obtained by estimating the areas of each section, just as in the upright position.

The theories of the *curve of flotation* and of the *cusps* were developed by *Dupin* in 1822, and by *Sir Philip Watts*[11] in 1871, but are far too intricate for general use.

If it so happens that the assumed *WL* does not cut off the correct displacement but is placed either too high or too low in the body plan, the error in the position of B_1 can be eliminated by using the Law of Moments on the excess layer and the displacement including this layer.

The C.G. of the layer is evidently at the C.G. of the inclined *WL*-plane. As a matter of fact, if the inclined *WL* is drawn as here shown, the error is usually quite negligible.

Although B_1 is nearly exactly located by the use of paper patterns, sometimes a still more accurate method is advisable, especially at the angle where the righting lever *GZ* crosses the basis, and perhaps at the angle of the maximum *GZ*. At any inclination, the wedge moment must be balanced by the displacement moment, but the wedge moment is no longer equal to the moment of inertia of the inclined *WL*-plane, as when the inclination is very small. However, by the use of the half-angle of inclination, originated by the author, the wedge moment is found to equal the moment of inertia of the half-angle *WL*-plane. By the half-angle is meant one-half of the actual angle of inclination, as for instance 15° for the 30°-inclination. The new method is based on *Guldinus' Rule* that any volume of revolution (whole or part) is equal to the area of the generator plane multiplied by the travel of its center of gravity.

Thus in Fig. 15, the generator plane is the half-angle *WL*-plane HH_1, of which there are two parts, one emerged, one immersed. The areas and the C.G.'s are easily calculated from the half-breadths at each section in the body plan. Let the generator plane revolve through the angle a, about the axis O_1 where the upright and the inclined *WL*'s cross. From O_1 draw short circular arcs at each section with a radius equal to the half-breadths of the half-angle *WL* HH_1. We thus get one emerged and one immersed circular sector at each section of the body plan, the areas almost exactly the same as the actual wedge section areas, no matter what shape the sections have.

Call the radius of the circular arc y, then at each section, the volume of the wedge per foot length $= \frac{1}{2} y^2 \times a$ the moment of the wedge $= \frac{1}{2} y^2 \times a \times \frac{2}{3} y \times \cos \frac{1}{2}a = \frac{1}{3} y^3 \times a \times \cos \frac{1}{2}a$ per foot.

The angle a is in circular measure $= a$ in degrees divided by 180, or $a°/180$, but $\cos \frac{1}{2}a$ is of course in degrees. At the given angle of inclination, both values of a are constant, and if all the different wedge moments are added by Simpson's Rule or by integration, the total wedge moment takes the familiar form

$$\tfrac{1}{3}y^3 \times L\ (a \times \cos \tfrac{1}{2}a) \tag{20}$$

where the factors to the left of the parentheses denote the moment of inertia of one side of the half-angle *WL*-plane. The moments of inertia

of both sides are easily estimated with the help of Fig. 13 (note parallel body), added together, and multiplied by $a \times \cos \frac{1}{2}a$. The moments are measured parallel to the inclined *WL*, hence the factor $\cos \frac{1}{2}a$. The volume of both wedges must be equal, and is estimated by multiplying the area of the *WL*-plane by the travel of its C.G. or

$$w = A \times ay_o \tag{21}$$

where A = the area, and ay_o the travel.

Formulas (20) and (21) are valid until the uppermost continuous deck touches the inclined *WL*, after which B_1 is best located by paper patterns, especially if there are partial superstructures.

Atwood's Formula. Stability at any angle of heel is measured by $D \times GZ$, termed the moment of statical stability. D = displacement of ship in tons, GZ = the horizontal righting arm in feet. $GZ = BB_1 \times \cos \frac{1}{2}a - BG \times \sin a$ (Fig. 15), and the first term

$$= \frac{\text{the horizontal wedge moment}}{\text{displacement}},$$

$$D \times GZ = D\left(\frac{\text{wedge moment}}{W} - BG \times \sin a\right) \tag{22}$$

This is the original Atwood's Formula, developed in 1798, but its drawback has always been the tremendous difficulty of estimating correctly the wedge moment. In our case this is done by Fig. 13 and formula (20). Note that although the volume of the wedges are equal, the *moments of inertia are unequal.* The point O_1 lies in the C.G. of the inclined *WL* if all estimates are correctly done.

Stability in Damaged Condition. If water should get into any compartment of a ship, it causes the vessel to sink deeper, to alter trim, and to lose stability. The mean sinkage and the trim will be the same as if a weight equal to the amount of the incoming water was added to the ship, except as to its effect on the area and the moment of inertia of the water plane. The mean sinkage if the water fills the compartment, is equal to

$$\frac{35 \times w}{\text{net area } WL\text{-plane}} \quad (\textit{see} \text{ formula 19})$$

If the water does not fill the compartment, cargo being present, then the sinkage is only a percentage of the value above, that is the percentage that the water actually occupies, termed permeability. For instance, a cargo of wood might almost fill the compartment, permeability 10%, whereas a cargo of heavy ore might increase permeability to 80%. The average is 63% for ships not carrying passengers below upper deck.

Sinkage being known (assumed parallel) the trim is calculated by formula (20), where wd = moment of incoming water to the C.G. of the original *LWL*-plane. Usually M_1G is not altered by the sinkage, since, although BM_1 is shortened, B is always raised. Displacement D and G remain the same, hence formula (19) is practically correct, even in the damaged ship.

The question might be asked how large a compartment may be before the vessel is in danger of sinking. This is termed subdivision but refers only to passenger ships over 150 G.T. and carrying more than 12 passengers. However, many cargo ships are now subdivided too, for additional security. Subdivision is treated in Part III-A more completely. (*See* pages 80–81.)

Free Water in Hold. If the water should not fill an empty compartment, it is termed *free water*, a source of danger not only to the stability of the ship but also to the strength, when rolling among waves. It makes no difference if the vessel be damaged or not, the danger is there just the same.

A deep tank partially filled may be taken as an example. As the ship rolls, the free water surface tends to stay horizontal but cannot, and the sloshing about increases the danger of capsizal. However, when estimating its effect, it is usually assumed that the free water surface remains horizontal at every inclination. The water in the tank behaves like a ship when heeled over, hence its metacenter rises above b as $\frac{i}{w}$, or $bm = \frac{i}{w}$, where the symbols have the same meaning as for the ship except that w = volume of water in tank.

When the vessel heels over to an angle a, the center of buoyancy b in the tank moves to b_1, causing the C.G. of ship and water to move from G to G_1 in the same direction as b to b_1. From the law of moments, $GG_1\ (W + w) = bb_1 \times w$, hence $GG_1 = \frac{bb_1 \times w}{W + w}$. If a vertical line is

drawn from G_1 in the body plan, it crosses the ship's center line at G_2 (*see* Fig. 16) which is the virtual C.G. of the ship plus the water in the vessel. Should G_2 rise to the vicinity of M, the ship's metacenter, the vessel capsizes. It thus becomes most important to find the position of G_2.

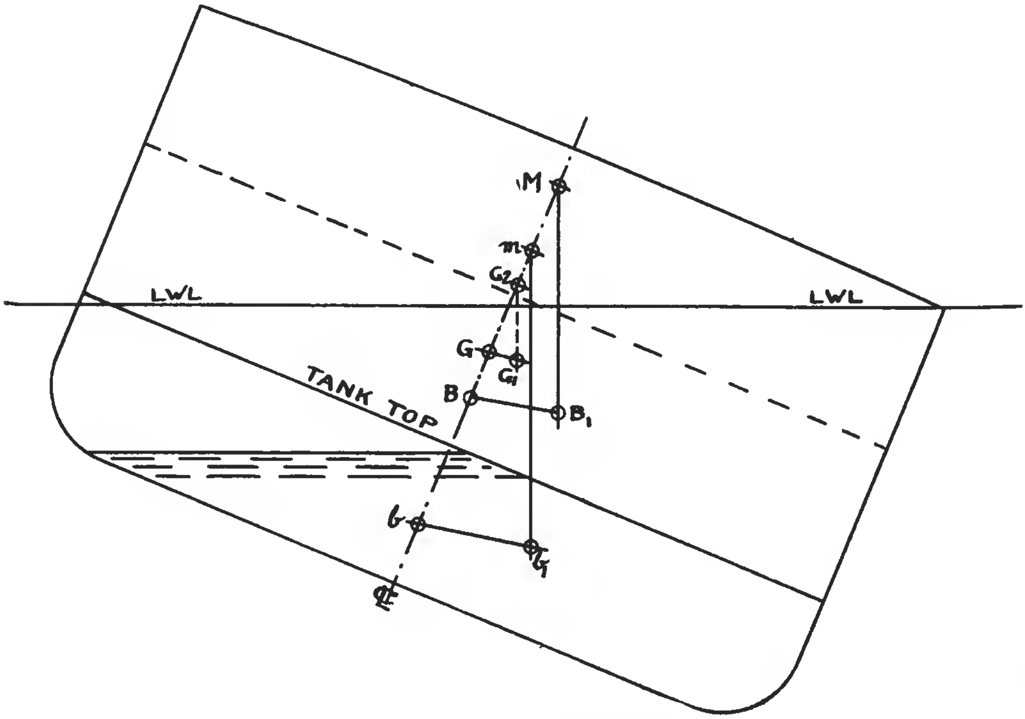


FIG. 16. FREE WATER IN SHIP.

The direction in which b and G travels is not parallel to the water surface but makes an angle $= \frac{1}{2}a$ with the latter, nearly. For all practical purposes, $GG_1 \times \cos \frac{1}{2}a$ equals the righting arm at the angle a in degrees. $GG_1 \times \cos \frac{1}{2}a = GG_2 \times \sin a$. Likewise, $bb_1 \times \cos \frac{1}{2}a = bm \times \sin a$. As shown above, $bm = \dfrac{i}{w}$, and $bb_1 = \dfrac{i}{w} \times \dfrac{\sin a}{\cos \frac{1}{2}a}$. When $\cos \frac{1}{2}a$ is eliminated,

$$GG_1 = \frac{i}{w} \times \frac{w \times \sin a}{W + w} = \frac{i}{W + w} \times \sin a,$$

and $$GG_2 = \frac{GG_1}{\sin a} = \frac{i}{W + w} \tag{23}$$

Note that G_2 is not an actual center of gravity like G and G_1, but only

a virtual center. Since MG is sometimes very small, perhaps only one foot, formula (23) shows that very little free water might cause a considerable list if not a capsizal.

BONJEAN'S CURVES. To facilitate trim and launching calculations, Bonjean's Curves are very useful. The areas of all sections in the body plan, from the keel upwards, are estimated *WL* by *WL*, and set off to scale from each section in the profile. At each *WL* the area below it is set off horizontally as length. At the *LWL* the area of each section is of course the same as that set off in the curve of areas, the same scale being used in both cases. The Bonjean's Curves are nearly straight lines, so that only one or two spots need be fixed between the keel and the *LWL*.

Now whenever the displacement of the ship at a certain trim is wanted, simply draw a line in the profile to represent the trim *WL*, read off the area at each section where the line crosses the section, and compute the displacement and the center of buoyancy for the trimmed condition. The mean draft and corresponding displacement are sometimes obtained from the curve of displacement, but it is never equal to the trimmed displacement and tells nothing about the position of the center of buoyancy that should fall in the plumb line from C.G. of the ship. Conversely, the C.G. can be located longitudinally at any trim by the use of the Bonjean's Curves. Sometimes curves showing the change in displacement for a trim of one foot either forward or aft, are handy, likewise curves showing the effect of trim on transverse stability. There are practically no limits to the number of curves that can be drawn to represent some special condition of the design, most of them just so much loss of time.

Bonjean's Curves are very useful for estimates of *floodable lengths* and *subdivision of ships*, as developed in Part III.

Part III-A
Naval Architecture As A Quasi-Science

NEW TYPE DESTROYER, U.S.S. *Dewey*, SHOWING VERY THIN BOW WAVE BUT A TREMENDOUS WAKE (ALL LOSS). SPEED-LENGTH RATIO, 1.98. *Official U. S. Navy photograph.*

THE STUDENT is now advancing into the region of Science as taught in textbooks and colleges with pure guesses as premises—for want of a better definition, here termed quasi-science as contrasted to the exact science as disclosed in Part III-B. In either case, an elementary knowledge of differentials and integrals would be a great help toward understanding.

FLOODABLE LENGTHS

It was shown in Part II, page 73, how free water in the hold of a ship either lowered its stability to the point of capsizing, or might cause the ship to sink if a large space were flooded. How great a volume of water is required to cause disaster depends on the stability and the freeboard of the vessel. To guard its stability, longitudinal bulkheads are built into all deep fuel or water tanks, greatly reducing their injurious effect. To guard against sinking, transverse bulkheads subdivide the ship into smaller compartments, the maximum lengths of which are termed *floodable lengths*. A vessel that can stand the flooding of one, two, or three such compartments is called a one-, two-, or three-compartment ship, and naturally the last one is the safest. The floodable lengths and the subdivision are now regulated by the rules of the *International Convention on Safety of Life at Sea*, ratified by most countries, and by the United States June 19, 1936. The rules apply to passenger vessels of 150 gross tons or over, carrying more than 12 passengers on foreign and coastwise voyages. The rules are called *Load Line Regulations*, and are obtainable at the Government Printing Office, Washington, D. C., to which the student is referred. The most important definitions cover the floodable lengths, the margin line, and permeability. In a vessel with a continuous bulkhead deck, the *floodable length* is the maximum portion of the length of the vessel that can be flooded without the vessel being submerged beyond the margin line. The *margin line* is a line drawn parallel to the bulkhead deck at the side and three inches below the upper surface of that deck. *Permeability* of a space is the percentage of that space which can be occupied by water.

The flooded lengths are again regulated by the *permissible lengths*

obtained from the floodable lengths by multiplying by the factor of subdivision. No permissible length shall exceed 1/5 of the ship's length or 80 ft., whichever is the less. This is the maximum spacing of the bulkheads for subdivision except the forepeak, which must not be longer than 5% of the vessel's length plus 10 ft., nor shorter than the 5%. The minimum spacing of the bulkheads is 2% of L plus 10 ft.

An unexpected effect of permissible length is that subdivision draft may invalidate freeboard draft in case the former is less. This applies only to passenger ships.

FACTOR OF SUBDIVISION. The factor of subdivision varies with the length of the ship and the nature of its service. Its numerical values extend from the factor A for primarily cargo ships, where $L = 430$ feet and over,

$$A = \frac{190}{L - 198} + 0.18$$

to the factor B for primarily passenger ships, where $L = 260$ feet and over,

$$B = \frac{100}{L - 138} + 0.18$$

CRITERION OF SERVICE. For a vessel of given length the appropriate factor of subdivision shall be determined by the *criterion numeral* as obtained by the following formulas,

$$C_s = \frac{M + 2P_1}{V + P_1 - P} \times 72 \text{ (when } P_1 \text{ is greater than } P)$$

and in other cases, $$C_s = \frac{M + 2P}{V} \times 72$$

C_s = criterion numeral
L = length of vessel on subdivision load line
M = the volume of the machinery space, plus the volume of any oil fuel bunkers situated above the inner bottom outside of the machinery space
P = the whole volume of the passenger spaces below the margin line

V = the whole volume of the vessel below the margin line
N = the number of passengers for which the vessel is certified
$P_1 = 0.6LN$

RULES FOR SUBDIVISION. The factor A shall govern the subdivision aft of the forepeak of vessels 430 ft. in length and upwards, and having a C_s of 23 or less. The factor B shall govern those vessels having a C_s of 123 or over. Those vessels having a C_s between 23 and 123 shall be governed by the factor F from the formula

$$F = A - \frac{(A - B) \times (C_s - 23)}{100}$$

The subdivision abaft the forepeak of vessels between 430 and 260 ft. in length having a criterion numeral

$$S = \frac{4691 - 10L}{17}$$

shall be governed by the factor 1.00. Of those having a C_s of 123 or more, by the factor B above. Of those having a C_s between S and 123 by the factor F,

$$F = 1 - \frac{(1 - B)(C_s - S)}{123 - S}$$

The subdivision abaft the forepeak of vessels less than 430 ft. but not less than 260 ft. in length and having a C_s less than S, and of all vessels less than 260 ft. in length shall be governed by the factor 1.00, and these provisions shall apply also to vessels of whatever length certified to carry over 12 passengers, but not exceeding $L^2/7000$ or 50, whichever is less.

There are many special rules concerning subdivisions for which the student should consult the government regulations.

PERMEABILITY. In determining the floodable lengths, a uniform average permeability shall be used for each of the following portions below the margin line:

1) The machinery space as defined in the subdivision rules
2) The portion forward of the machinery space
3) The portion aft of the machinery space

For steam vessels the average permeability throughout the machinery space shall be determined by the formula

$$80 + 12.5 \frac{(a - c)}{v}$$

a = volume of passenger spaces below the margin line within the limits of the machinery space
c = volume of 'tween deck spaces below the margin line within the limits of the machinery space which are used for cargo, coal, or stores
v = whole volume of the machinery space below the margin line

For vessels propelled by internal combustion engines, the uniform average permeability shall be taken as 5 greater than that given by the above formula.

The uniform average permeability throughout the portion of the vessel before (or abaft) the machinery space shall be determined from the formula:

$$63 + 35 \frac{a}{v}$$

a = volume of passenger spaces situated below the margin line before (or abaft) the machinery space,
v = whole volume of the portion of the vessel below the margin line before (or abaft) the machinery space.

Note. The volumes here defined shall be the molded volumes below the margin line down to the top of the keel.

Calculation of Floodable Lengths. The calculations are made and the results plotted on the *profile* of the ship. Start with drawing a straight line upwards from the ends of the subdivision load line slanting at an angle of 63.5° with the base line towards amidships. The subdivision load line cannot come above the highest freeboard mark, because that would be illegal. In winter, the vessel cannot be loaded deeper than to the lowest freeboard mark for the same reason. Next, draw the margin line 3 inches below the bulkhead deck at the side, and estimate the whole molded volume and its center of buoyancy up to several trim lines, touching the margin line in different places. Here the Bonjean's curves become a great help, since the areas of all

sections up to the trim line can be read off directly, and treated according to Simpson's Rule. One such trim line should touch the margin line (be a tangent to it) at each machinery bulkhead, because the permeability is different on each side of these bulkheads. Other places on the margin line should be taken about 15% of the load line length from each end of the vessel, and a few intermediate places.

To obtain the floodable length at each place, two different methods can be used. One was developed by Professor J. J. Welch for the British Board of Trade[12] from a number of standard designs of different lengths, the floodable lengths of which could be transferred to similar ships as ordinates to the floodable length curve. This method requires a complete set of the Welch Diagrams, and besides, the form of ships has changed so much since 1915 as to make the results of little value for the modern ship designer. The other method estimates directly the floodable lengths for each design as well as the permissible lengths at each place along the margin line.

The idea is, generally, to find by trial and error the flooded volume that would cause the ship to sink to the margin line. The volume divided by the mean sectional area at the place in question equals the floodable length there for 100% permeability. This length divided by the estimated permeability in decimals, say 0.65, and multiplied by the factor of subdivision (*see* above) finally gives the permissible length of the compartment. The curve of permissible (or floodable) length is plotted by erecting ordinates at each selected point of the margin line and setting off the length at the same scale as the profile of the vessel.

The volumes of the displacements at the subdivision load line and at the trim line being known, their difference equals the volume causing the sinkage from one line to the other,

$$t = \frac{w}{\text{net area of margin line plane}} \qquad (24)$$

t = sinkage that is measured off from the profile
w = difference in the volumes of the displacements

Since both these factors in formula (24) are known, the net area of the margin line plane (practically the same as the bulkhead deck) can be estimated, as well as the approximate length of the compartment. If this length multiplied by the average sectional area below the selected place on the margin line, equals w corrected for permeability,

that is to say $w/0.65$ above, we have only to test the moments of the volumes as a check of the computations.

The center of buoyancy B for the undamaged ship, as well as B_1 for the trimmed displacement are both known. The center of buoyancy b for the flooded compartment is found most quickly by balancing a paper pattern cut from the curve of areas of the trim line displacement (*see* Fig. 17). Then, if W = volume of displacement of the undamaged ship,

$$W \times BB_1 = w \times B_1b \tag{25}$$

in order that B_1 shall fall in the same vertical line as G_1, the new center of gravity of the flooded ship. Formula (25) is not strictly accurate, since the vertical line BG is not parallel to B_1G_1, but the error is negligible. If formula (25) checks with our computation, the length l of the compartment is set up from the molded base line through the spot X on the margin line to mark a point of the floodable length curve of the vessel. For constructive reasons, X is supposed to be in the middle of the compartment and not at b, and the length of any floodable compartment is obtained by drawing slant lines from the corresponding point on the floodable length curve. The slant is 63.5° to the base line, the tan 63.5° = 2.00. Sometimes the vertical scale is made only one-half of the length scale, in which case the slant angle is 45°. Since b is never at the middle of the compartment except in the middle body, this makes a slight error in the volume of the flooded compartment, but always on the safe side because the volume moment of the floodable compartment is actually slightly less than that assumed in the calculations.

After the curve of floodable lengths has been laid down, the factors of subdivision A for cargo ships and B for passenger vessels, determines the permissible lengths of each compartment. If the factor is greater than 0.50, the vessel is a one-compartment ship, if over 0.333 up to 0.50, a two-compartment ship, and if over 0.25 up to 0.333, a three-compartment ship. For instance, our design with a length of 326 ft. would have a factor of subdivision equal to 1.66 as cargo ship, and 0.71 as passenger vessel, and in both cases would be a one-compartment ship, provided its criterion numeral were 84.2 or less. For passenger vessels, two-compartments start at a length of 450 ft., and three-compartments at 805 ft.

LOSS OF STABILITY IN DAMAGED CONDITION

Every ship sustains a loss of stability when trimmed, on account of her wedge-shaped ends, the water planes of which have very low moments of inertia. For instance, a parabola has less than half the moment of inertia of the rectangle of equal dimensions, although its area is two-thirds of the latter. But the greatest danger to the ship occurs

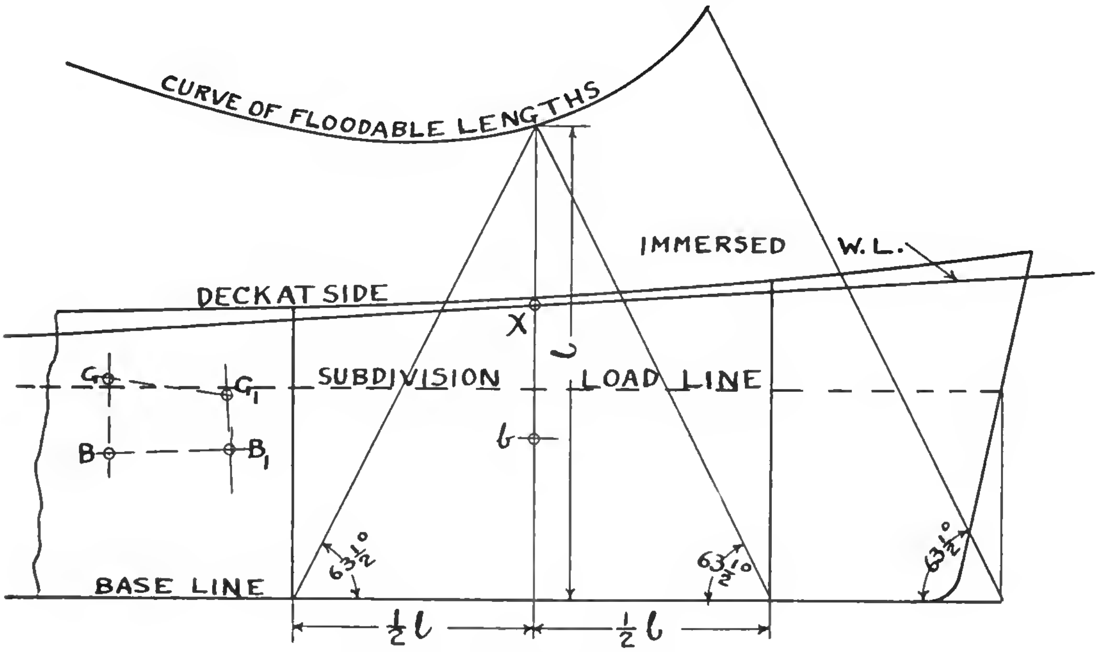


FIG. 17. FLOODABLE LENGTHS CURVE.

while a damaged portion is slowly filling with water. The higher the permeability and the larger the compartment, the greater the danger. This danger is always present in a one-compartment ship rolling among waves, and should be investigated by assuming different levels of water inside the compartment, each with an angle of heel to just immerse the margin line. When estimating B_1 and M_1 of the partially flooded condition, the trim water plane is of course intact, although the incoming water not only increases the displacement but also lowers G to G_1, the new center of gravity of the ship. Methods for estimating have already been given, but permeability should be taken into account, say 0.80 for the machinery space, and 0.63 for fore-and-aft spaces. There is likewise a similar loss in the inside water plane moment of inertia, a loss that decreases with the height of the water level to zero, when the compartment is filled.

Wing compartments are another source of danger when damaged only on one side. The Load Line Regulations of the United States allow a maximum heel of 7°, in this case in the worst service condition anticipated. There are many other rules that must be followed in regard to subdivision, with which the ship designer must be familiar.

ROLLING

Rolling is so closely related to stability and to floodable lengths that it will be considered here, although its theory demands an understanding of integrals. The mathematical theory of rolling was first investigated by W. Froude in 1861,[13] although several unsuccessful attempts had been made earlier. Rolling is produced by wave action on a ship when the wave crests run parallel to the vessel's center line, just as pitching is caused by waves coming head on to the ship. There is also heaving when the waves lift the ship, and this movement is most unpleasant to the passengers. All wave movements create additional stresses in the ship's hull, that force the shipbuilder to increase its weight many times more than would be necessary in smooth water.

The Froude theory of rolling considers first rolling in still water when no resistance is affecting the roll. Evidently a vessel cannot roll in still water, hence movable weight must be used to start the rolling. Once started, it would continue to swing about a longitudinal axis very near the center of gravity G of the ship. The period of rolling is the time in seconds for a double swing from one side to the other and back again.

$$T = \pi \times \sqrt{\frac{k^2}{MG \times g}} \times 2 \tag{26}$$

π = 3.14, the ratio between circumference and diameter of a circle
k = the radius of gyration of the ship about the axis of rotation
MG = the metacentric height
g = the acceleration of gravity, 32.16 ft. per second
k is an imaginary quantity such as $W \times k^2 = I$
W = displacement in tons of 35 cu. ft.
I = mass moment of inertia of the ship about the longitudinal axis.

Formula (26) has only theoretical value, as it is nearly impossible

to calculate k with an acceptable degree of accuracy. Besides, the formula renders far too high values when MG is small; for instance, t = infinity when $MG = 0$. However, I can be obtained as well as T from rolling experiments, at least for medium size ships, as for instance the American steamer *America*, $T = 28.0$ seconds,[14] and especially for warships, as, for instance, the British battleship, *Revenge*, $T = 16.8$ seconds,[15] because naval vessels have a large crew that can be easily augmented for the rolling experiments, from which I is deduced as follows:

$$I = \frac{T^2 \times W \times GZ}{4\pi^2}$$

$GZ = MG \times \sin a = MG \times a$ for small angles,

$$I = \frac{T^2 \times W \times MG \times a}{4\pi^2} \tag{27}$$

From many British experiments it has been found that k is nearly proportional to the beam B of a ship $\left(\text{for battleships } k = \frac{B}{3}\right)$, *see* reference 2, page 205, and formula (26) finally becomes

$$T = \frac{C \times B}{\sqrt{MG}} \tag{28}$$

where C is a constant that varies only between 0.40 and 0.45, the former value for warships, the latter for merchant ships.

Example. The rolling period of our design,

$$T = \frac{0.45 \times 52.5}{\sqrt{3.26}} = 13.1 \text{ seconds.}$$

Rolling among waves has been largely studied, and a highly intricate theory has been developed, of little practical value except that it has shown that the period is almost the same whether resistance is encountered or not. When a wave is approaching a ship broadside on, its familiar sloping surface lifts the nearest side of the vessel, thus giving its masts an inclination not only to the vertical but to the wave slope as well. If it so happens that the wave period equals the natural period of rolling for the ship, the inclination of the latter would be doubled

by each wave until the ship might turn bottom up. It is reported of some old frigates that they rolled their main yards under when anchored in open roadsteads.

The theory of waves (of which more later) teaches that the period as well as the speed of a wave is proportional to the length of the wave, speed in knots, length in feet,

and
$$\left.\begin{aligned} T_w &= 0.442\sqrt{l_w} \\ V_w &= 1.341\sqrt{l_w} \end{aligned}\right\} \qquad (29)$$

The length of the dangerous waves for our design,

$$l_w = \left(\frac{13.1}{0.442}\right)^2 = 800 \text{ feet, from crest to crest.}$$

Such long waves are common only in the North Atlantic in winter, and explains the extra freeboard of 2 inches for ships under 330 ft. traversing the North Atlantic in winter.

Luckily for the safety of our ships, ocean waves are never very regular in shape, but consist of all lengths mixed together, with the longest wave as a foundation. Particularly, ocean waves never display the wide, straight crests assumed by pure theory and by some artists, but are criss-crossed in every direction. Hence their effect on the ship, although bad enough, is never as hazardous as suggested by the theory of rolling.

Means to Reduce Rolling. A moving ship sets in motion great masses of water clinging to its surface, whether the motion is ahead or simply rolling, or a combination of both. By adding long fins to the surface, rolling is reduced but, alas, the resistance of the vessel is increased at the same time. Such fins are termed bilge keels because placed at the turn of the bilge, and like bar keels at the bottom of the ship, are very efficient against heavy rolling. Bilge keels are now added to every seagoing ship, the deeper the better, but constructual reasons limit their depth. Evidently, the more and the deeper the bilge keels, the more water would cling to the surface, to be set in motion at every roll, and in opposite directions. It takes a great force to do this, a force that must be supplied by the ship, and that naturally tends to reduce rolling. Ships have seldom more than two bilge keels, one at each side, but very big ships are fitted with docking keels under the bottom that also help to reduce excessive rolling.

Many other means have been tried against rolling, such as anti-rolling tanks, gyroscopes, stabilizing fins, but all have been found wanting. Submarines were at one time stabilized by side fins, but these were found too fragile for practical use.

ROLLING OF TYPICAL SHIPS. If still water is divided by imaginary horizontal and vertical lines, the former follow the wave profile in its undulations, the latter move to and fro as do grain stalks when struck by gusts. In both cases the movement decreases with the depth below the surface of the water. Clearly, a wide, flat board must follow the profile and roll *with the wave*, whereas a narrow deep board must move to and fro as the wave passes, and always *against the wave*, but rolling very little at that. For reasons of stability and of resistance, an ocean-going ship is neither flat nor deep, hence its motion among waves is intermediate but always worse when the period of rolling of the ship equals the wave period as already stated.

Form of midship section influences rolling. As between the circular arc and the rectangle, the former offers hardly any resistance to rolling, hence would cause exceedingly heavy oscillations, as noted in some destroyers and torpedo boats. The rectangular form is almost as steady in water as on land, but in neither case is the form of the midship section continuous throughout the length of the ship, and thin, wedge-like ends offer great resistance to rolling. *Sails* also have a steadying influence, perhaps best shown in the modern racing yacht, that would roll excessively but for its sails, on account of its nearly circular sections, all fore and aft.

EQUATION OF ROLLING MOTION. The equation of rolling motion, as first developed by W. Froude,[13] rests on the fundamental axiom of mechanics that force equals mass multiplied by acceleration. In a rotary motion such as the rolling of a ship, the mass is its mass moment of inertia, the acceleration is angular $= \frac{d^2\theta}{dt^2}$ in differential symbols, and the force is the moment of stability of the rolling ship. In formula (26), mass $= \frac{W}{g}$ and mass moment $= \frac{W}{g} \times k^2$, hence

$$-W \times GZ = \frac{W}{g} \times k^2 \times \frac{d^2\theta}{d^2t} \tag{30}$$

The minus sign is employed because the moment of stability is opposing the rotary motion. For small angles of heel, the metacentrum of the vessel is a constant point, and $GZ = MG \times \theta$—*see* formula (27). The differential equation (30) thus becomes

$$\frac{d^2\theta}{dt^2} + \frac{MG \times g}{k^2} \times \theta = 0$$

the solution of which under certain assumptions, leads to formula (26) for unresisted rolling in still water.

Froude in 1861, and Professor Rankine in 1872,[16] tried to develop formulas for rolling among waves, and Froude even gave a method of graphic integration,[17] but beyond showing the influence of synchronism of wave and ship periods, and of the probable effect of resistance to rolling, their efforts were unsuccessful as regards the most important question—the necessary range of stability to guard against dangerous rolling. Or, as Rankine puts it, "the only practical way of providing against the effect of . . . rolling is to allow a margin in the limiting angle of stability, whose proper amount can be determined by experience alone."

MODEL EXPERIMENTS

The use of models for solving complicated ship problems was first suggested by the great Sir Isaac Newton, the discoverer of the Law of Gravitation, and probably by Leonardo da Vinci 200 years earlier. But Newton was the first to state the theoretical connection between the model and its full-size *precedent*, called by him the Principle of Similitude. His principal discovery was that the speeds of the model and of the *precedent* should be proportioned to the square roots of their dimensions.

However, Newton's principle was quickly forgotten, and when the Dutch at the height of their naval power made model tests, the models were towed *at the same speed* as the ships that they represented. Naturally the water could not close in aft of the model at such high relative speeds, unless the afterbody were given great length and fineness, resulting in the Cod's Head and Mackerel Tail ship form that held sway for hundreds of years.

William Froude took hold of the Newton idea of models but called it the Law of Comparison when applied to ships. He stated that when

the speeds of two ships, or of a model and its precedent, were proportional to the square roots of their lengths, their resistances were proportional to their displacements. But Froude soon found out that this was not true, and seems to have been the first to understand the dual nature of ship resistance. In 1871 he persuaded the British Admiralty to build a model experimental tank at Torquay, and began there his famous experiments on skin frictional resistance of planks up to 50 ft. long, painted or covered with various materials.

But right at the start of his experiments he made a serious error: he submerged his planks entirely, thus making the results applicable only to submarines, and not at all to models for whose resistance *surface tension* plays a big part. At the time, nothing was known of Laminar Flow discovered by Osborne Reynolds in 1883, but not applied to ships until 1900.[18] Another error was Froude's assumption that the skin resistance of curved surfaces of ships were similar to that of flat planks. W. P. Roop[18] proved that Laminar Flow was present in models up to a speed of 3 knots when the experiments were evaluated by the Liljegren formulas for frictional skin resistance. The formulas that Froude, and later Tideman, developed from the Froude experiments were thus fundamentally incorrect, and not valid above a plank length of 50 ft. for speeds above 3 knots. Yet these formulas were accepted not only for ship models but also for ships up to 1,000 ft. in length.

Fig. 18 shows the big differences between the Tideman and the Liljegren curves for a 10-ft. long model at speeds up to 8 knots, corresponding to 80 knots for a 1,000-ft. ship. The curves are plotted for the ratios $\frac{R_f}{AV^2}$, termed specific skin resistance.[18]

R_f = frictional resistance in pounds
A = wetted surface in square feet
V = speed of model through water in knots

If R_f were proportional to V^2, the curves would be straight horizontal lines, but the diminishing ordinates at higher speeds were interpreted by Froude and others to mean a lesser speed exponent than 2, which is only seemingly true as far as it goes. The Liljegren curve shows the extent of Laminar Flow to be 3.51 knots for a 10-ft. model. The undulations represent the struggle of the water particles to acquire turbulent flow even at low speeds. The spot marked E.M.B. marks a value obtained by the Experimental Model Basin in Washington,

D. C., and is far above the Tideman curve although a little lower than the Liljegren curve. The reason for this will be mentioned presently.

It was not until Dr. Kempf in Hamburg made his famous experiments with 250-ft. long pontoons and pipes[19] that correct formulas for skin resistance could be made to emerge from the maze of contradictory evidence. On one point, however, all these experiments were in perfect agreement: the resistance was greatest at the very cutwater

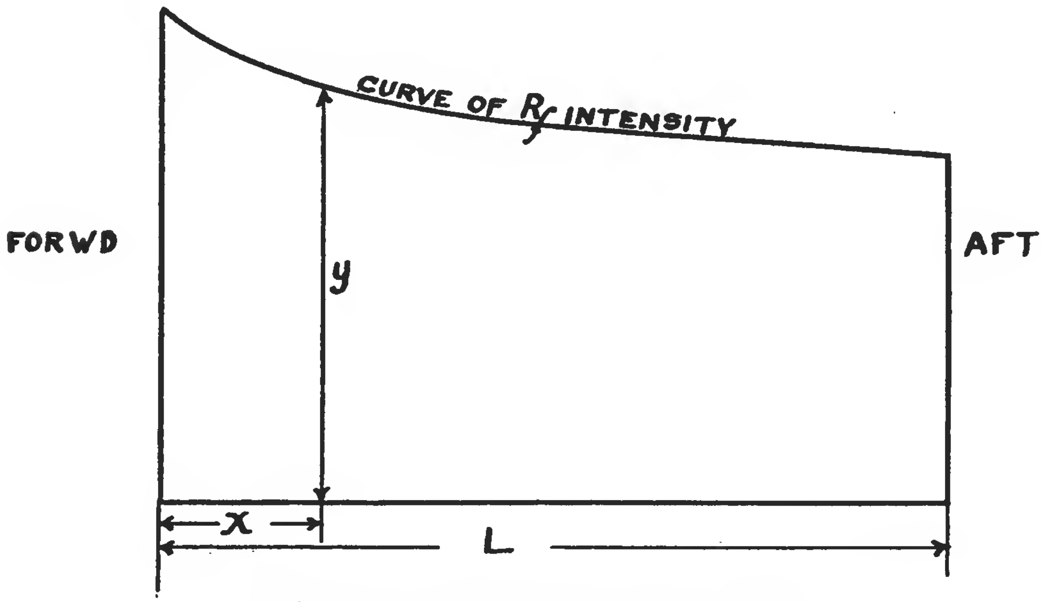


FIG. 19. SKIN RESISTANCE INTENSITY CURVE.

of the model, its intensity per square foot diminishing with the distance from the cutwater until it at last became constant in value. Fig. 19 depicts the resulting curve where x = distance from cutwater, and y = intensity. Attempts to find the equation of y led only to the Froude, Gebers, and other formulas, with the exponent for y less than 2. But all through the realm of physics, resistance has been proven to vary as V^2, not as $V^{1.89}$ or some number less than 2. Here was a conundrum. It was solved as in Fig. 20 by drawing a line parallel to the base, through the lowest spot of the intensity curve, dividing the intensity into two parts, one of constant value y_2 over the entire length of the plank, and one of diminishing value y_1.

The skin resistance of the constant part $= 2y_2 \times L \times G$, where G is the immersed depth of the plank, and L its length. But the added resistance of the shaded part spread over the entire surface of the plank was not so easily deduced, and right here was where the old formulas

failed. On the small length dx, the skin resistance $= 2y_1 \times G \times dx$, and summation over the entire L gives the skin resistance of the shaded area $= 2\int_0^L y_1 \times G \times dx$. The total skin resistance

$$R_f = 2y_2 \times LG + 2G\int_0^L y_1\, dx \tag{31}$$

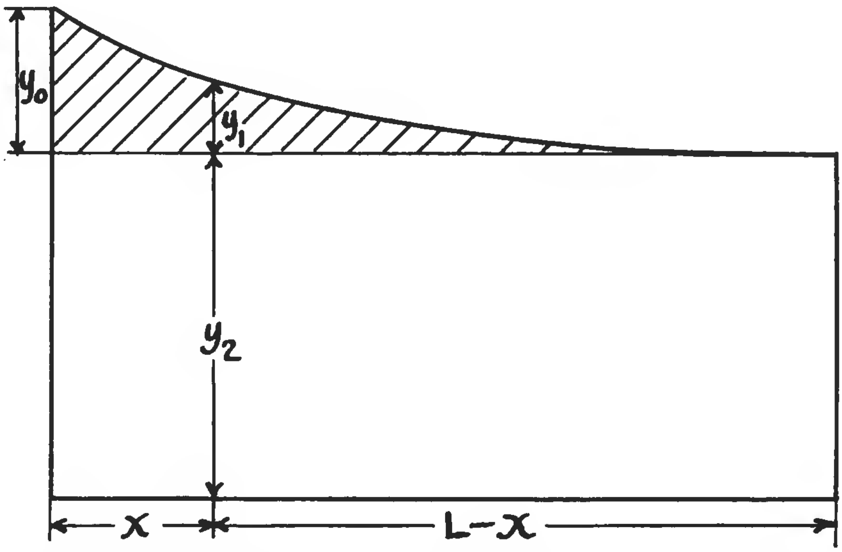


FIG. 20. SKIN RESISTANCE INTENSITY SEPARATED INTO TWO COMPONENTS.

2 LG is the wetted surface of the rectangular plank, but in case G varies with distance from cutwater x, as in ship forms, formula (31) takes on a different aspect altogether.

The intensity y in Fig. 19, equals R_f/AV^2 as stated above, and the same applies to y_1 and y_2 proportionately. But y_2 is constant, hence $R_f/AV^2 = \text{constant} = c_2$. This is true for all surfaces of whatever shape or contour. But in order to integrate $\int_0^L y_1\, dx$, a connection must be found for y_1 and x, as, for instance,

$$y_1^n = C(L - x)$$

where n is the exponent of the intensity curve. But the resulting integral is exceedingly complex, and the matter must be approached in another way. The shaded area in Fig. 20 representing the excess re-

sistance caused by the inertia of the water particles set in motion by the plank surface,

$$\int_0^L y_1 \; dx = \frac{y_0 L}{n+1}$$

according to well-known methods of mensuration, y_0 being the value of the y at the cutwater. In Fig. 20 it is assumed that the excess resistance covers the entire length of the plank, but experiments by Kempf[20] and Gebers[21] have definitely proved that its extent is quite limited and dependent on the speed of the plank and on the roughness of its surface. Kempf experimented with 37-mm. (1.5-in.) brass pipes, one part of which was movably attached to the tank carriage, with other parts of different lengths fixed ahead of the former. He found that the specific skin resistance R_f/AV^2 became constant at 10 ft. and more from the head of the pipes. His findings were confirmed both by his experiments with 350-mm. (14-in.) pipes,[20] and his pontoon.[19] But above all, his small pipe data, generously supplied to the writer, enabled the latter to evaluate n from his curves (*see* Fig. 21). Kempf here made a serious error in that he placed the values of R_f/AV^2 at the middle of the pipe length instead of at the center of skin friction which is ahead of the middle. When this error was corrected, the writer found $n = 4$, and the excess resistance area in Fig. 20 thus $= 1/5\ hl$, if h = height and l = length. It was further found from Fig. 21 that y_0 is proportionate to V, and the extent of the excess resistance proportionate to $V^{1/4}$, which simplified the finding of the correct formula. Excess resistance then equals

$$2GV \times V^{1/4} \times \text{a constant } c_1$$

and the whole skin frictional resistance,

$$R_f = AV^2\left(\frac{c_1}{LV^{3/4}} + c_2\right) \tag{32}$$

A perusal of this formula shows that instead of the Reynolds value $\frac{1}{VL}$ for pipes comes $\frac{1}{LV^{3/4}}$ for planks. For the small speeds of the Reynolds' experiments the difference is negligible but increases with the speed. It is also evident from formula 32 that the term involving

C_1 becomes negligible for $VL = 1.000$ or more, after which the specific resistance is constant.

The values of c_1 and c_2 were obtained from Gebers' experiments with 5-meter planks,[21] perhaps the most accurate experiments of their kind. The speeds ranged from 3.88 knots to 14.54 knots, thus outside of Laminar Flow, and the values of R_f/AV^2 for the limiting speeds were 0.00828 and 0.00710.

$$1)\ 0.00828 = \frac{c_1}{16.4 \times 3.88^{3/4}} + c_2$$

$$2)\ 0.00710 = \frac{c_1}{16.4 \times 14.54^{3/4}} + c_2$$

Eliminating c_2 by subtraction, $c_1 = 0.0852$ which, substituted in (2), gives $c_2 = 0.00641$, both values for varnished smooth surfaces of rectangular planks. How closely these values approached those of Gebers. is seen from the following table.

TABLE V

GEBERS' SKIN RESISTANCE COEFFICIENTS

Values of R_f/AV^2

Speed knots	3.88	5.82	7.76	9.70	11.64	13.58	14.54
R_f/AV^2 experiment	828	778	754	733	721	714	710
R_f/AV^2 formula	828	778	752	735	723	714	710

Note. The decimal zeros have been omitted for simplicity, the actual values being 0.00828, etc.

EDGE EFFECT. Wood planks cannot be made very thin but must have a certain thickness to prevent undulations or whipping action that would increase their resistances. Hence the edges make some resistance that must be deducted in the formulas, and this applies both to vertical and longitudinal edges. In case of fully submerged planks such as Froude's, all four edges are effective, but with Gebers' 5-meter planks, only three edges need be taken into account. However, there are too many unknown factors that make direct estimate of the edge effect impossible but, by assuming it to amount to a certain percent of the total R_f, values can be found that make the sum of errors a minimum. For instance, in Table V the sum of errors was only 3.1 for an assumed edge effect of 2.5%, and both higher and lower percentages

gave greater errors. For 5%, the sum of errors became 8.7, and so forth, hence 2.5% was chosen. Deducting 2.5% from the end values of R_f/AV^2 in Table V, and solving for C_1 and C_2 as before, the former = 0.0830, and the latter = 0.00625, and the corrected formula (32),

$$R_f = AV^2\left(\frac{0.0830}{LV^{3/4}} + 0.00625\right) \tag{33}$$

So far only rectangular planks have been considered. In case of ship models with curved surfaces, an adjustment must be made in formula (33) for the longitudinal curvature. The transverse curvature is included in the girth of every section. Just as in planks, the skin resistance of curved forms can be separated into two parts, one constant in intensity over the entire surface, the other highest at the cutwater and diminishing aft. But the girth G is no longer constant as in planks, which makes the difference.

Let $G \times ds$ represent the surface element affected by skin resistance, the resistance of the element $= 2y_2 \times Gds \times \cos a$, where a is the angle between the element and the line of motion or the center plane of the model. Cos a is introduced because only the longitudinal component is resisting the motion—a fact always overlooked hitherto. Now $ds \times \cos a = dx$, and the total resistance of the constant intensity

$$= 2\int_0 y_2 \times G \times dx$$

Y_2 is assumed constant $= c \times V^2$, and $2\int_0^L G \times dx =$ the wetted surface A of the model, projected on the center plane, hence the entire integral $= c_2 \times AV^2$, exactly similar to that of a plank as found in formula (31).

As regards the excess resistance near the cutwater, it at first seemed to admit only graphical solutions—too cumbersome for ship estimates. Ultimately, an analytical solution was obtained even in this case.

In Fig. 22, the area $ABCD$ denotes the *effective* wetted surface of a model of the length L and the girth G (one side). The intensity line at any point x must follow the contour curve ABC. The zone of excess resistance extends over the length l_z, and from the similarity of the intensity lines and the contour curve, also a length l_z beyond the deepest point B of the curve. The length of the lines of equal intensity

(vertical in a rectangular plane) follow the contour line forward, and evidently equals the length of the contour curve from A to B, and the summation of all the flow lines add up to the rectangle with the sides l_z and G.

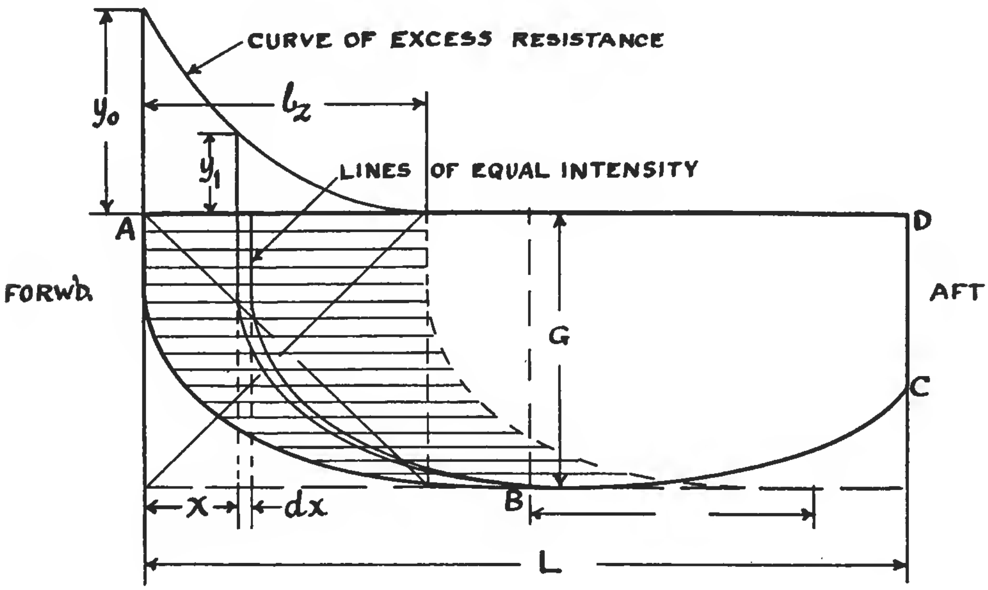


FIG. 22. EXCESS SKIN RESISTANCE.

The excess skin resistance thus $= 2G\int_0^L y_1\,dx$, exactly the same as in formula (31) for rectangular plane. THIS IS THE MOST FAR-REACHING DISCOVERY EVER MADE IN THE WHOLE THEORY OF SKIN RESISTANCE, and makes correct estimates of model skin resistance possible, without recourse to graphical solutions. There is only one limitation: l_z must not much exceed $L/2$, otherwise there is a loss in excess resistance, although not serious until $l_z = L$ or above.

It is to be noted that the wetted surface A is not equal to $2LG$ but to $2fLG$, where f is the wetted surface coefficient,

$$f = \frac{A}{2LG}$$

If inserted in formula 33,

$$R_f = AV^2\left(\frac{0.0830}{fLV^{3/4}} + 0.00625\right) \tag{34}$$

which is the formula for estimating skin resistance of any ship model at any speed below that where $l_z = L$, Fig. 22.

There remains now to find expressions for l_z and for y_0. Kempf's experiments[20] have made it possible to evaluate y_0 in pounds per square foot,

$$y_0 = 0.0637V \tag{35}$$

From Gebers' experiments[21] it was apparent that l_z varied very little with the speed, as the fourth root in fact, and in feet,

$$l_z = 6.57\sqrt[4]{V} \tag{36}$$

The length of the initial zone l_z is practically the same for every surface material except paraffin. As the constant c_1 in the formulas is obtained directly from the experiments, the exact value of the coefficient in (36) is not important.

In the case of l_z being very much longer than L, a graphical solution must be used, but it can happen only with small models at excessively high speeds, such as hydroplanes.

SKIN RESISTANCE OF SHIPS

For $LV^{3/4} = 1.000$, or more, the term in formula (34) involving $c_1 = 0.0830$, becomes negligible as has been mentioned, and the skin resistance varies as AV^2. All models are tested in fresh water, and ships run generally in salt water. Nor is that all, for ship surfaces are very much rougher than the varnished wood used in models. For steel surfaces painted smooth, Kempf[19] states that the skin resistance is 4% higher than for wood, but leaves no means to decide how c_1 and c_2 vary. By analyzing Gebers' experiments,[21] it was found that while c_2 varied 4%, c_1 increased as 1.04^2 or 8%. For steel surfaces in salt water, $c_2 = 0.00625 \times 1.04 \times 1.025 = 0.00665$, $c_1 = 0.0830 \times 1.08 \times 1.025 = 0.0928$. These two values inserted in formula (34) show the skin resistance of ships with smooth, welded surfaces as now common. For ordinary lapped and riveted steel surfaces, Kempf[19] fitted laps on his 250-ft. pontoon, which increased the value of c_2 to 0.00690 for a length of 250 ft. Since plates cannot be rolled long enough for big ships, c_2 increases slightly for lengths over 250 ft., as for instance, U.S.S. *Saratoga*, for which $c_2 = 0.00700$, only a slight increase.

Skin Resistance of Fouled Bottoms. A ship out of dry dock, quickly gathers slime, shells, and other marine growth on her surface by which her resistance is increased. There is no safe rule by which to measure this increase, except that it is much larger in the tropics than in the arctics, and that growth gathered in salt water quickly dies in fresh water. The average increase can be taken as 33% after six months.

Minimum Wetted Surface. For a fixed displacement and midship area, there is a certain ratio B/D that offers the smallest wetted surface, and hence the smallest skin resistance per ton of displacement. The mathematical expression for WS is

$$2\int_0^L G\,ds$$

the integral taken over the entire length. Because of the curvatures of the ship's surface, the integral cannot be solved. It has been stated that for the purpose of estimating skin resistance, the integral becomes

$2G\int_0^L dx$, and can be solved.

The girth at any section is a function of beam and draft at the section, or $g = C(b/2 + d)$, where C is a constant. C remains surprisingly constant over the entire surface but usually $C = 1.0$ at the ends of LWL. In the early stages of design, C can be estimated from the midship section, $C = \dfrac{G}{B/2 + D}$.

G is the girth of one side, B the beam, and D the draft of the midship section, the area of which is assumed constant $= C_x \times BD$. The

integral of WS $= 2\int_0^L (b/2 + d)\,C\,dx = C\int_0^L b\;dx + C\int_0^L 2d\;dx.$

But $\int b\;dx$ = area of LWL, and $\int 2d\;dx$ = twice the area of the longitudinal center plane of the vessel. Hence

$$WS = C(c_0B + 2c_1D)L \tag{37}$$

Since $D = \dfrac{A_x}{C_xB}$, the variable terms $= c_0B + 2c_1\dfrac{A_x}{C_xB}$, and is a mini-

mum when $c_0 - \frac{2c_1A_x}{C_xB^2} = 0$. Multiply by B, $c_0 = 2c_1 \frac{A_x}{C_xB}$, and this transfers to

$$\text{for minimum WS, } c_0B = 2\,c_1D \tag{38}$$

Note: A_x is the area of the midship section.

Formula (38) is valid for any form of immersed body. For instance, a solid of rotation with a parabola as generating curve must have $2/3B = 2(2/3D)$, or $B = 2D$, whereas if the keel is straight as in a ship, $2/3B = 2D$, or $B = 3D$, if its wetted surface is to be a minimum.

The constant C in formula (37) varies slightly with the form of the body. In the solid of rotation, $C = 0.785$, whatever the generating curve is like. For the Taylor Standard Series, the prismatic coefficient $= 0.555$, $C = 0.827$. In a parallel midship body, c_0 and $c_1 = 1.0$, $C = \frac{G}{B/2 + D}$ always. This constant does not alter the ratio B/D in formula (38) but in a modern ship it is dependent on the length of the midship body. For instance, if the Taylor Standard Series is given a middle body equal to $L/2$, C would $= \frac{0.827 + 0.900}{2} = 0.864$, 0.900 being the ratio $\frac{G}{B/2 + D}$ for the Series. From the Taylor formula for WS proportionate to $\sqrt{L \times W}$, C would $= 0.882$, evidently too high a value. The latter formula is useless for ships with parallel middle body.

For a preliminary design, C is always fixed for the midship section, and the sum of the end values can be taken to equal 1.5 for a square stem and a cruiser stern. If a section at $1/4L$ from each end is sketched in, and their girth constants measured, then by the use of the Simpson's Rule, the WS constant is obtained with great accuracy, but of course will be checked, for future use, when the design is finished. No addition is made for the obliquity of the water lines, but the rudder should be included, as its area increases c_2 in formula (38). Other appendages are added separately, as their resistance ratios are much greater in the model than in the ship, and all are affected by pressure resistance as well as skin resistance. A *bulbous bow* adds little to the skin resistance except as its girth is increased, but it may add tremendously to the pressure on the forebody at high speeds when the bow lifts, and when

the vessel pitches badly in a headsea. The bulbous bow is supposed to add some of the advantages of the torpedo to the ship, but the torpedo is a submerged vessel, and has no hollow lines fore and aft, whereas the bulbous ship is a surface vessel, and has got extremely hollow lines that add to its resistance.

APPENDAGE RESISTANCE. Hitherto there seems to have been no reliable measurements of the appendage resistance of merchant vessels, although G. S. Baker[22] gives some data for twin screw ships with rudder, bilge keels, shaft struts or spectacle bossings, presumably obtained from models without corrections for the full size ships. The data appears as ratios between the appendage resistance and the total resistance at 7 speed-length ratios as follows,

$V/\sqrt{L}$	0.40	0.50	0.60	0.70	0.80	0.90	1.00
$\frac{\text{Appendage } R}{\text{Total } R}$	0.188	0.138	0.115	0.105	0.101	0.100	0.100

The steadily decreasing ratios are probably correct for 10-ft. models, because only the total resistances are compared. How near these ratios represent full size conditions at corresponding speeds, is not possible to state. Taylor[23] guessed at ratios of 0.20 for warships with bilge and docking keels, and perhaps two pairs of struts on each side. For twin screws, Italian destroyers[24] with rudder, struts, shafting, and bilge keels, the ratios were given as follows, in knots and feet,

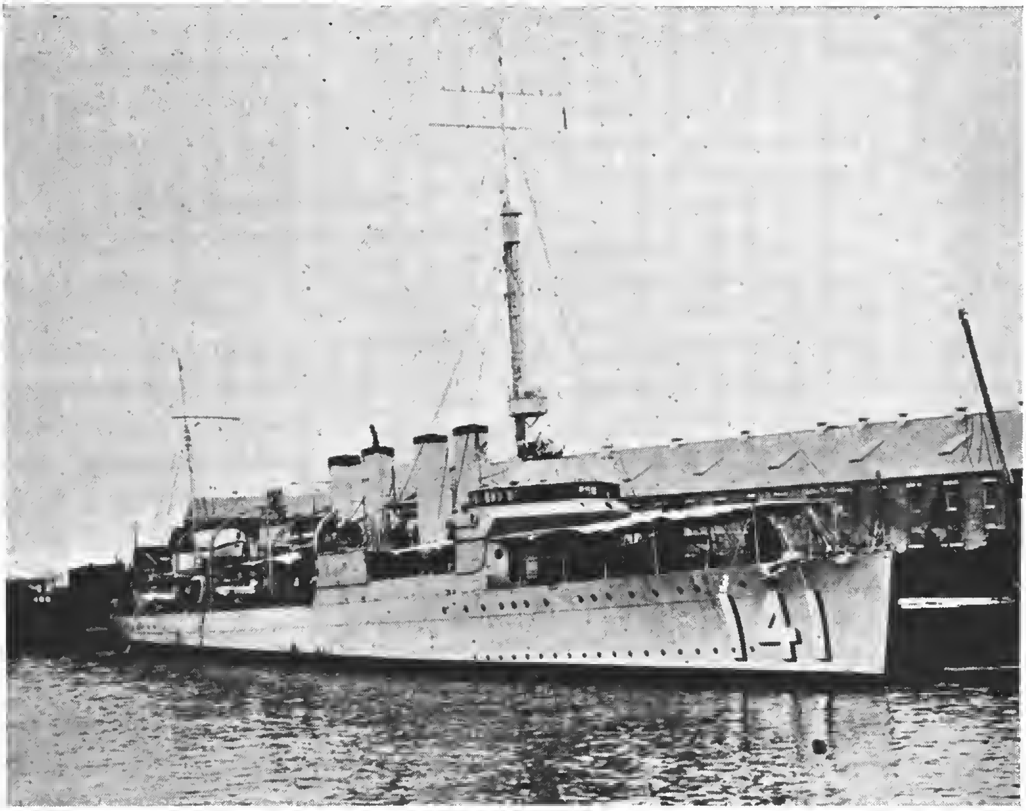
$V/\sqrt{L}$	1.105	1.380	1.660	1.933	2.210
$\frac{\text{Appendage } R}{\text{Total } R}$	0.111	0.115	0.107	0.107	0.102

As regards these ratios, the reference is evidently to models, because no destroyer can be towed at a $\frac{V}{\sqrt{L}}$ of 2.21 equal to a speed of 38 knots.

Through the courtesies of the Experimental Model Basin in Washington, D. C., the writer was supplied with full data of appendage resistance of *U.S.S. Hamilton*,[25] a twin-screw destroyer, and *U.S.S. North Carolina*,[26] a twin-screw heavy cruiser, both sets of tests made by Captain H. E. Saunders, U.S.N. The *Hamilton* was tested both in full size and in a 20-ft. model, the *North Carolina* in models of four different sizes. While all models over 16 ft. suffered great resistance increases from the effect of the basin walls and bottoms, the writer has been able to discount this effect, and to obtain the correct model resistances

in each case. The work is not finished at this writing, but a comparison has been made with the Italian destroyers cited above, and *U.S.S. Hamilton*, which was fitted with rudder, bilge keels, and shaft bossings and one pair of struts on each side. For the *Hamilton:*

$V/\sqrt{L}$	1.105	1.380	1.660	1.933	2.040
$\frac{\text{Appendage } R}{\text{Total } R}$	0.165	0.114	0.077	0.091	0.083



OLD TYPE DESTROYER, U.S.S. *Hamilton*. SPEED-LENGTH RATIO, 2.04. *Official U. S. Navy photograph.*

The *Hamilton* experiments with and without appendages fitted to the model gave perfectly accurate spots for the curves of resistance based on speed. At the highest speeds the wall effect reduced the appendage ratios of the model quite appreciably, over 10%, and the ratios would be near 0.100. At the $V/\sqrt{L} = 1.66$ the bow wave interference also reduced the ratio appendage $R \div$ total R.

It should be mentioned that all destroyers suffer from very high resistances per ton of their displacements at the lowest speed-length ratio. Twin screw merchant ships with lower resistance per ton, actu-

ally have higher relative appendage resistance than destroyer models. For instance, the *Hamilton* model at $V/\sqrt{L} = 0.905$ suffers a total resistance of 23.8 lbs. per ton, whereas the 20-ft. model of the Ferris Liner[7] had only 18.1 lbs. per ton although fitted with quadruple screws and bossings. Actually the appendage resistance of *Hamilton* amounted to 5.03 lbs. per ton of displacement, whereas the Ferris Liner showed only 3.13 lbs./ton. At the same model speed, the *Hamilton* appendage resistance was 1.65 lbs., that of the Ferris Liner 1.71 lbs.

These values are for 20-ft. models. For ships at corresponding speeds the appendage resistance ratio is much smaller because the skin friction of the model appendage is nearly inversely proportionate to the length of the appendage, and not to the length of the model. A bilge keel of half the length of the model increases its specific resistance about 50%, *see* formula (34), shorter appendages such as shaft bossings and struts much more. Part of the appendages are affected by the boundary layer of water set in motion by the model but the thickness of the layer is very diminutive, and its mean speed only one-sixth of the model's,[21] hence its effect on resistance can be disregarded. But the *direction* of the water particles aft of the midship body is no longer parallel to the ship's line of motion and must be taken into account because any new angle of incidence means a new cutwater and excess resistance, as well as an increase in the total pressure resistance. In every case it is only the longitudinal component that affects the resistance, as stated.

Before proceeding, a few common mistakes in the theories of ship resistance might be stated here:

1) The boundary layer does not increase in thickness from forward aft along a plank. Whoever has rowed in a fast 8-oared shell, or sailed in a long, narrow racing yacht, can attest to this fact.

2) In a ship form, the boundary layer remains constant near the surface of the ship, but part of the water particles lose contact with the skin as the girth grows less and less aft of the midship body.

3) No streamline action can take place in a partly immersed vessel, hence the pressures and the speeds are constant over the entire wetted surface except where the bow waves alter them. Gebers[21] has proved that the speed over the surface is just one-half of the actual speed of the model but a little less than one-half if the surface is rough. The higher the retardation, the higher the specific skin resistance.

4) All theories of resistance based on streamline action are valid only for submerged vessels. It is even a moot question how far the body must be submerged before streamlines can act but, judging from the swimming of seals, it must be several times deeper than its greatest transverse dimension.

Such mistakes are, of course, fatal to any theory, but are too common in all pioneer work. Compare the crates used as the first airplanes, or the horseless carriages that were the first automobiles. The curious ideas of Gebers and many others[27] who assumed that skin resistance was inversely proportional to some function of L, are directly traceable to mistakes (1) and (2). The assumption leads to the untenable result, as W. P. Roop[18] has shown, that the specific resistance approaches zero at large values of VL, speed times length. As Roop[28] also proved, all formulas for skin resistance based on VL lead to a scattering of spots in any diagram of frictional resistance. On the other hand, diagrams based on the writer's $LV^{3/4}$ do not scatter the spots at all, and are the final proof of the correctness of formulas (33) and (34). As these formulas are based on the most accurate model experiments known, nothing else could be expected.

Laminar Flow. As now conducted, model experiments give the total resistance of any model with almost incredible accuracy. The results could be extended—converted as it is termed—to full-size ships were it not for *laminar flow* at low model speeds, and for *wall effect* at high model speeds. For 20-ft. models, the former ceases to be felt at a speed of 3.5 knots, and the latter is not felt until a speed of 5.0 knots. This leaves a range of only 1.5 knots within which accurate conversion from model to ship can be made at present.

Laminar flow was subject to experiments and analyses[18] by W. P. Roop who supplied the writer with complete data for developing a formula similar to (33). Owing to the resurgence or the undulations of the laminar flow resistance, its formula has not been completed to date, and perhaps no single formula can be expected to do the work; but the analysis is continued. Incidentally, the Roop experiments have shown the futility of all theories hitherto developed of laminar flow—and they are legion. To name only one, von Kármán[29] states that the resistance from laminar flow varies as the 3/2 power of the velocity, whereas the writer's very minute analysis of the Roop experiments of the 10-ft. model proved most decidedly that the same resistance varies as the square of the velocity, or as V^2, up to a speed of 1.0 knots. Above this speed, the resurgence becomes more effective, the Kármán formulas still more futile. Further, the Roop experiments entirely disproved the transition stage idea, since the curve joining the laminar and the turbulent flows on the diagrams, is a double-curved or sinuous line not at all like the theoretical line.

In lieu of formulas, Fig. 18, page 97, gives a clear picture of the specific resistance R_f/AV^2 within the range of laminar as well as of turbulent flow, of the resurgence, and of the joining curve. Fig. 23 does the same things for the 20-ft. models. In both instances, the wetted surface coefficients = 0.80, *see* formula (34), and a small correction is necessary for other values of f. To make this correction for a given speed, deduct the value of c_2 = (0.00625) from the specific resistance shown by the curves, increase or decrease the remainder as the actual f is smaller or greater than 0.80, and add again 0.00625.

EXAMPLE. For a 10-ft. model at a speed of 2.25 knots, with $f = 0.85$, Fig. 18 shows $\dfrac{R_f}{AV^2} = 0.01000$. Deduct 0.00625, multiply remainder 0.00376 by the ratio of the two f's, or by $0.80/0.85 = 0.00354$. Add 0.00625, and the correct value of $R_f/AV^2 = 0.00978$. Having thus obtained the correct value, we get the skin or frictional resistance of the model by multiplying with AV^2. Deducting it from the total resistance as found by model experiments, leaves the pressure resistance that can be directly converted to the ship according to Newton's Law of Similitude.

WALL EFFECT. Before the conversion from model to ship can be reliable, there still remains the possibility that the model resistance has been unduly increased by the wall effect. The 10-ft. model is usually free from wall effect except in very small model basins, but the 20-ft. model begins to feel the walls at a speed of 5.0 knots if the immersed midship section of the model amounts to more than 0.45% of the water section of the basin. Clearly, the wall effect can be best ascertained by towing models of the same vessel but of different lengths, and luckily the Experimental Model Basin, Washington, D. C., has done just that, in two instances, with three models of the *U.S.S. Saratoga*, and with four models of the *U.S.S. North Carolina*. Through the courtesies of Captain E. F. Eggert, and of Captain H. E. Saunders,[25] complemented with the trial trip data of the *Saratoga* by Comdr. J. T. Alexander, U.S.N.,[28] it has been possible to the writer to measure the wall effect on the 20 ft. models of the two ships mentioned above. Any method used is dependent for accuracy on the correct formula for skin resistance, *see* formula (34), since otherwise the pressure resistances of the different models cannot be ascertained with exactness.

Considering the nature of the pressure resistance, it became clear that at least two factors influenced the wall effect. One was the speed-length ratio $V/\sqrt{L}$, the other was the ratio between the midship section area of the model and the cross section of the basin water. The area ratio for the *Saratoga* = 0.00243, and that for the *North Carolina* = 0.00740, hence it would seem that the latter should suffer more from wall effect. This was true, but the difference was not in proportion to their area ratios, probably on account of the very superior hull form of the *North Carolina*.

The 3 models of the *Saratoga* had a length of 20.0, 28.33, and 32.0 ft., respectively. After plotting the pressure resistances for several $V/\sqrt{L}$, on the basis of model lengths, all the curves agreed that a model of 15 ft. in length was not affected by the basin walls. The speed-length ratios used were 0.783, 1.110, 1.185, and 1.390 for all three models. For a 15-ft. model, the lowest $V/\sqrt{L}$ means a speed of 3.04 knots, well inside the laminar flow range, but this fact did not alter the result, at least not appreciably.

For each $V/\sqrt{L}$ the 15-ft. model length gave a distinct and minimum value of the pressure resistance per ton of displacement. The plotted curves showed the increase for the longer models, the reversal of which indicated the wall effect on the 20-ft. model. For instance, at $V/\sqrt{L}$ = 1.39, the increase in pressure resistance between the 15-ft. and the 20-ft. model was 1.124, the reversal = 0.890 indicated the reduction due to the wall effect of the measured pressure resistance of the 20-ft. model. Likewise for $V/\sqrt{L}$ = 0.783, the reversal was 0.975. These two reversals belonged to the limiting speed of the *Saratoga* model, and furnish two values of the *reduction coefficient* for the wall effect formula. Models less than 15 ft. suffer nothing from wall effect in the Experimental Model Basin, Washington, D. C.

The wall effect formula must include $V/\sqrt{L}$ and the area ratio as already mentioned. After many attempts, the best form of the formula was found to be,

$$C_w = 1 - C(V/\sqrt{L})^n \times R_a$$

C_w = wall effect reduction coefficient
C = a constant obtained from experiments on 20-ft. models
$V/\sqrt{L}$ = speed-length ratio
n = the exponent or index of $V/\sqrt{L}$, also obtained from models
R_a = area ratio between the immersed midship section of the 20-ft. model and the cross section of the basin water

Of these symbols, only C and n are unknown. If the others are given values from the model experiments,

1) for $V/\sqrt{L} = 0.783$, using the *Saratoga* model,

$$0.975 = 1 - C \times 0.783^n \times 0.00243$$

2) for $V/\sqrt{L} = 1.39$,

$$0.890 = 1 - C \times 1.390^n \times 0.00243.$$

Transforming and dividing (2) by (1),

$$4.40 = 1.775^n, \text{ and } \log 4.40 = n \times \log 1.775,$$
$$0.6435 = n \times 0.2490, \text{ from which } n = 2.58.$$

This value of n is so near 2.50 that the latter is used. There remains now only C, easily obtained from (2),

$$0.890 = 1 - C \times 1.39^{2.50} \times 0.00243,\ C = 19.90.$$

So far we have assumed that R_a is directly active in increasing the wall effect, which must be tested by using values from *North Carolina* in the formula. At $V/\sqrt{L} = 1.110$, the model experiments give a wall effect reversal $= 0.905$,

$$C_w = 1 - 19.90 \times 1.110^{2.50} \times 0.0074$$
$$C_w = 1 - 0.191 = 0.809.$$

This value, being far below 0.905, proves that some function of R_a must be introduced. Of many different exponents tried, the square root of R_a gave the most accurate results. The constant C in (1) and (2) is altered as $\dfrac{0.00243}{\sqrt{0.00243}} = 0.0492$, and $C = 0.980$. Hence,

$$C_w = 1 - 0.980(V/\sqrt{L})^{2.5} \times \sqrt{R_a} \qquad (39)$$

For the *North Carolina*, formula (39) gives $C_w = 0.891$ within 1.5% of the actual value. For the *Saratoga* at the same $V/\sqrt{l} = 1.110$, $C_w = 0.938$, from the formula, the actual value $= 0.940$.

The 15-ft. model length gave no indication of wall effect. Its area ratio $R_a = \left(\dfrac{15}{20}\right)^2 \times 0.00243 = 0.00137$ which might be accepted as the maximum R_a that can be used in any model basin without errors resulting from wall effect.

The result attained by the experiments with models of different lengths shows that all tables and formulas of pressure resistance are erroneous if the models tested have an area ratio R_a greater than 0.00137. This is particularly the case of the Taylor Tables since the Standard Series possess a $R_a = 0.00708$, almost five times the maximum allowable. The tables need correction not only for frictional resistance but also for pressure resistance, the aggregate error reaching 50% in many cases. Corrected curves will be pictured later in the book (*see* Fig. 25, page 116).

Air Resistance. It is generally assumed that a towed model is not affected by air resistance but a moment's reflection will show that the hollow model suffers an appreciable increase of resistance unless covered with a light deck. It is probable that the air resistance of the model nearly equals the 2.5% of the water resistance estimated for ships in which case no addition for air resistance should be made to the ship's converted resistance.

Thousands of experiments have proved that air resistance is proportional to the area affected, and to the square of the speed,

$$R\text{-air} = CAV^2$$

But since the air velocity never is constant over the surface of a model, recourse has been had to towing ship models *upside down* in water. Unfortunately the water resistance of the model is never proportional to V^2 (as in air), besides which laminar flow and pressure resistance increase the differences. Such experiments can have only a relative value but still have furnished some useful facts, such as that rounded corners of deck houses reduce resistance.

Most noteworthy experiments in water were made by G. Hughes[30] who also tested a 17-ft. model of the *Mauretania* in air, on a high tower—only the part of the ship above water, of course. This test gave wide variations between the air and the water resistances, even when the different densities are taken into account, and no reliable coefficient could be derived therefrom. The most important result was that the air resistance is maximum when the angle of incidence is 35°, the angle of burble or stalling in aircraft practice, and in rudder turning moments. For a ship of the Atlantic liner type, the increase is 41%, but for a cargo ship with many booms and derricks the increase is 30%, and for a tanker still less, only 11.5%.

The generally accepted formula for air resistance is

$$R\text{-air} = 0.0035AV^2 \tag{40}$$

where A is the area in square feet opposing the air velocity, and V as usual is the speed in knots. The resistance is in pounds. But the constants from reference 30 is 0.0044 for the cargo ships, and 0.0034 in the case of the liner. Some very useful experiments were made with the Great Lakes steamer *Greater Detroit*.[31] A strong head wind slowed the vessel 0.60 of one knot at a speed of 20 knots, and increased the engine power by 10% if the speed were constant. In other words, the air resistance amounted to 10% of the water resistance, whereas it is seldom more than 2.5% in a cargo ship. The lake vessel has very large superstructures.

Conversions from Model to Ship. If model experiments have been made for a proposed ship, it is necessary to first deduct any wall effect from the model pressure resistance by the use of formula (39). The pressure resistance is found by deducting the frictional resistance from the total resistance, and its amount is directly converted to the ship in the proportion of the displacement ratios between the model and the ship. The frictional resistance of the ship is estimated next, and added to the pressure resistance to give the total resistance. If no model experiments are available, the engine power can be interpolated at the given speed from Table IV or from diagrams in Part III-B. Should a ship with higher speed-length ratios be designed, one of those previously mentioned must be used if a vessel of the highest performance is wanted. The difference between the best $V/\sqrt{L}$ and the worst is over 50% of the pressure resistance as shown by many Diagrams of model experiments made by Froude,[32] Taylor,[23] and many others. Indeed, the Taylor Diagrams show a difference of over 100% instead of 50% even when corrected for frictional resistance—a beautiful example of wall effect at high speeds. Evidently the difference cannot amount to quite 100% owing to the slight internal friction among the water particles making up the bow waves.

All conversions rest on Newton's Principle of Similitude amplified by Froude's Law of Comparison, both valid for the model and its ship, but not from ship to ship on account of limitations from stability and draft. Both laws state that when speeds vary as the square roots of lengths, the resistances should vary as the displacements of the

model and of the ship. But, as already mentioned, the laws apply only to the pressure resistances when free from wall effect, and very few models are free from it. This fact accounts for the many incorrect conversions and engine power estimates seen in papers on resistance and powering of ship. Only by subterfuges such as *roughness allowances* can agreement sometimes be reached between model and ship, and then only in one spot. The surfaces of model and ship, while different in roughness, do not vary appreciably from model to model, or from ship to ship. And even if old steel plates get pitted, modern paints and painting methods largely make up for it.

If the dimensions of the ship are n times those of the model, their areas vary as n^2, and their volumes, weights, and displacements vary as n^3, and their stability moments as n^4—but right here their similitude ends. Neither their resistances nor their engine powers follow the Law of Similitude. It seems that William Froude was the first to discover these facts, and to make use of them in the interpretation of his model experiments. However, as Froude had no knowledge of laminar flow, of correct skin resistance, or of wall effect, his conversions from model to ship were altogether erroneous. All his model experiments were within the laminar flow range, and his larger models must have encountered wall effect in the small tank at his disposal. This is specially true of his most famous experiments with the model and the ship *Greyhound.*

The *Greyhound* was a wooden man o' war, coppered and fitted with auxiliary power as well as masts and sails. In the experiments the vessel was towed behind another warship, the *Active*, at various speeds up to 12 knots. The *Greyhound* was without masts and screw propeller, and was brought by ballast to any desired draft and trim. The model was made of paraffin, 10 ft. long, built to a scale of 1/16. The towing of the ship presented many difficulties that were overcome with genuine skill.[33] All attempts to obtain agreement between the model and the ship resistances have been futile to date. The writer has worked on this problem for ten years but it is only recently—with his discovery of the wall effect—that complete agreement can be hoped for. However, since ships are not coppered today, and models are seldom made of paraffin, the *Greyhound* experiments have only historical value. Many other model-ship towing experiments have since been made, all with the same lack of agreement. The full-scale thrust and engine

power trials of *U.S.S. Hamilton*[25] are especially worthy of attention.

Shallow-Water Resistance. We have seen how a model's resistance is influenced by wall effect. Something of the same kind is active in increasing the resistance and the engine power of ships in shallow water, and still more so in canals. For each speed there is a depth of water at which the resistance begins to increase, first slowly, then faster as the depth decreases. In shallow but wide, open waters, this depth is equal to one-half of the length of a trochoid wave of the same speed as the ship. The length of a trochoid wave $= V^2/1.8$ when V is measured in knots, hence minimum depth or

$$\text{Depth of water not influencing resistance} = \frac{V^2}{3.6} \tag{41}$$

For instance, at 6 knots, the *perilous depth* is 10 ft., a depth common in big rivers such as the Mississippi, where the speed should never exceed 6 knots or 7 miles per hour. But it is evident that the draft of the vessel also affects the resistance, since the deeper the draft in proportion to the depth of water, the more obstacles are put in the way of the water particles filling in the void after the vessel. There is no data available on this point except the rule of rivermen that 1 ft. is the minimum distance between the river bottom and the ship's bottom.

Sir Philip Watts[11] made extensive experiments on the British destroyer *Cossack* in waters of 444- and 240-ft. depth, and found excessive increase in resistance at the smaller depth, amounting to 50% at a speed of 21 knots. The perilous depth is 123 ft. at this speed which does not agree with formula (41). But there is no saying that the increase in resistance might not begin at 240 ft. in the case of the *Cossack*. Sir Philip also states that at the critical depth v^2/g, the increase in the resistance, is maximum at constant speed. The critical depth for 21 knots is 39.3 ft.; after this depth the resistance begins to decrease. The symbol v is here feet per second, $g = 32.16$.

Shallow-water waves are found to increase in both height and length over ocean waves (trochoidal). At a depth of $0.25l_w$ the length is increased 9%, and 52% at a depth of $0.125l_w$. The main effect of increased length is that the humps and the hollows in the resistance curves occur at a lower speed-length ratio, for instance, in the shallow water where *Cossack* was tested, the hump is at 21 knots, but in deep

water it would come at 23.2 knots, or at 10% higher speed. The length of the *Cossack* is not stated but, assuming it to be 270 ft., $V/\sqrt{L} = 1.28$ and 1.41 for the speeds mentioned.

Resistance in Canals. The resistance of a vessel in a canal is greatly increased, partly by the backward flow of the water past the ship, and partly by the wall effect. The backward flow is dependent on the speed of the ship overland, on the ratio between its midship area and the cross-sectional area of the canal, and on the sinkage of the water level in the canal opposite the ship. The sinkage has been estimated by Schaffran of the Berlin Model Basin,[24] using a flume built to scale of the Dortmund-Ems canal, and a suitable model barge. The barge itself was tested in the canal, and the conversion from model to barge gave almost perfect agreement. His theory is far too long and intricate to be quoted.

The wall effect has already been treated, formula (39), but the area ratios found in a canal are a hundred times greater than those used in a model basin. On the other hand, speed-length ratios are usually much smaller in a canal, and the wall effect not much different in percentage, in the canal and in the model basin.

In estimating the *backward flow* it is found that the sinkage of the water level in the canal does not influence the speed of the flow except to a limited degree. Let the symbols be,

V = speed of vessel overground in knots
V_1 = speed of backward flow in knots
A = area of the water cross section of the canal, in square feet
A_1 = net area of water section between the sides of the vessel and of the canal
A_2 = area of sinkage cross section of the canal—mean width of canal at undisturbed water level times sinkage in feet

Then,

$$\frac{V_1}{V} = \frac{A - A_1}{A_1},\ A_1 = A - A_x - A_2,\ (A_x = \text{midship section area})$$

$$A - A_1 = A_x - A_2, \text{ assume } A_2 = CA_x,\ (C = \text{a constant})$$

hence $A - A_1 = (1 - C) \times A_x$, and $A_1 = A - (1 - C) \times A_x$,

$$\frac{V_1}{V} = \frac{(1 - C) \times A_x}{A - (1 - C) \times A_x}$$

Divide both divisor and dividend by A_x,

$$\frac{V_1}{V} = \frac{1 - C}{A/A_x - (1 - C)} \tag{42}$$

Since $(1 - C)$ is included in divisor and dividend, it is clear that a small error in estimating C does not appreciably alter the ratio V_1/V. For assume as limiting values $A/A_x = 4.0$, and $C = 0.10$, then $V_1/V = \dfrac{0.90}{3.10} = 0.29$. Increase C by 50% to 0.15, $V_1/V = 0.27$, an increase of only 7.5%. Usually A_2 is very small compared to A_x and the error still less.

In estimating the *wall effect*, the pressure resistance of the barge is increased, and its value must be divided by C_w in formula (39). Assume $V/\sqrt{L} = 0.50$, and $R_a = 0.25$, $0.50^{2.5} = 0.177$, $\sqrt{R_a} = 0.50$, and $C_w = 1 - 0.98 \times 0.177 \times 0.50 = 0.903$ (approx.), which is not a high value of C_w. Its reversal $= 1.11$, an increase of pressure resistance of only 11%, but it must be remembered that R_p is very high in bluff-ended vessels like river barges. Formula (39), like all other formulas developed by the writer, is based on most careful experiments in model basins, and can always be relied on. The speed for estimating the frictional resistance in canals is $(V + V_1)$, as obtained from formula (42), but the increased velocity does not increase the pressure resistance since $V_1 = 0$ at the ends of the vessel, where R_p is originated.

Kempf[35, 36] has made extensive model experiments with canal and river vessels in his Hamburg model basin, both as regards resistance and screw propulsion. Professor Kenneth S. M. Davidson of Stevens Institute of Technology made self-propelled tests of a big river towboat that have been analyzed by the writer and found extremely interesting, but not yet made public property.

Available model test data might often be useful to ship designers if corrected for frictional resistance and for wall effect. Fig. 24 shows curves of corrections of R_f/AV^2 for 20-ft. and for 10-ft. models, the most usual sizes, when the Tideman formula was employed by the experimenter in estimating frictional resistance. The curve for the 20-ft. models applies especially to the Taylor[23] Standard Series with a prismatic coefficient $C_p = 0.555$, and a wetted surface coefficient $f = 0.80$, for which extensive diagrams have been prepared. When a value is to be corrected from the diagrams, first estimate the Tideman skin

frictional resistance, and increase by the correction factors shown in Fig. 24 for 20-ft. models. The pressure resistance is obtained from the diagrams and added to the R_f by the Tideman formula, which gives the total resistance. When the corrected R_f is deducted from the total R, the remainder is the corrected pressure resistance R_p, that may be as much as 50% lower than values obtained from the Taylor diagrams. Notice that the values of Tideman and of the writer's formulas for R_f/AV^2 are equal at the speed-length ratio of 1.80, and that above this point, the values are *lower in the writer's formula*. But, long before this, wall effect has made the Taylor diagrams thoroughly useless.

It has already been mentioned that the Taylor diagrams apply only to ships without parallel middle body, for which an increase in the displacement-length ratio can be had only by increasing beam or draft or both, since C_p is supposed to remain constant. If the dimensions are unaltered, and C_p increased by insertion of a parallel middle body, how would that affect the pressure resistance per ton of displacement? —On this point the Taylor diagrams leave no answer at all, except that the bow and stern angles of incidence are increased with C_p, and also the resistance per ton, although very slowly, since for constant dimensions the displacement varies with C_p. It must never be forgotten that the Taylor diagrams are based on a model length of 20 ft., and are quite useless for any other length, as seen from Fig. 24.

What then is the effect of a parallel middle body?—After considerable research extending over a decade or more, the writer is inclined to believe that the *total* resistance per ton of displacement is practically constant—no matter how long the middle body. The frictional resistance varies slower than displacement, and the pressure resistance faster, the net result being no or very little change, at least for speed-length ratios below 1.0. This result applies particularly to the favorable speed-length ratios 0.63, 0.72, 0.85, and 1.11.

For more accurate estimates of pressure resistance, the important question is the choice of the prismatic coefficient in the Taylor diagrams. For any given C_p and beam-draft ratio, an increase in displacement can only be had by increasing both beam and draft, regardless of the speed-length ratio. Since the length of the models of all Taylor diagrams is 20 ft., any increase of the displacement means a corresponding increase of the displacement-length ratio. In other words, a 25% addition to the displacement makes $\dfrac{W}{(0.01L)^3}$ exactly so much

larger. W = displacement in tons. The choice of C_p was limited to 0.555 because the design was printed in reference 23. The corrected Taylor diagram for $V/\sqrt{L} = 0.75$ is shown in Fig. 25 for B/D ratio of 2.92, the value chosen for the design. The original curves for $B/D = 2.25$, and for $B/D = 3.75$ are redrawn in Fig. 25, but based on displacement-length ratios for instant reference from any given design.

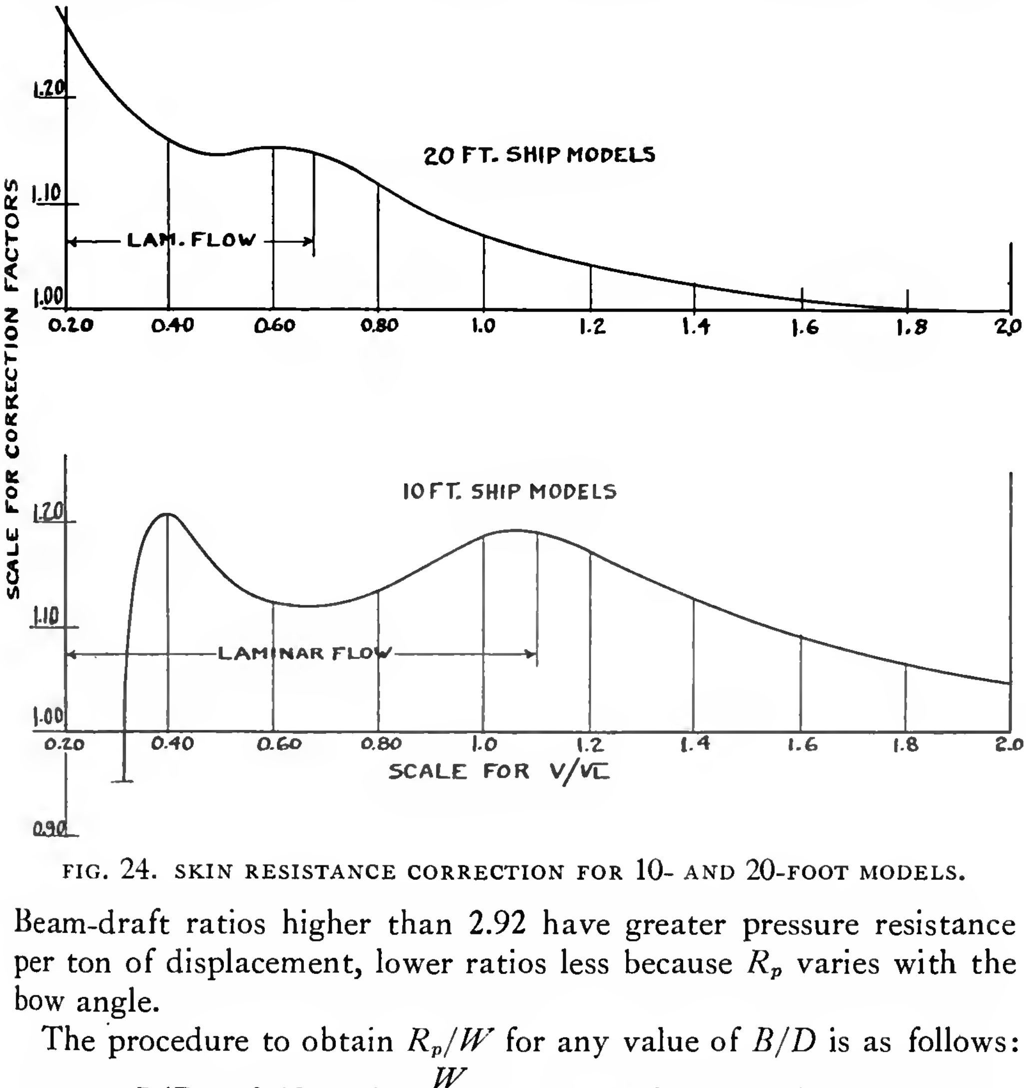


FIG. 24. SKIN RESISTANCE CORRECTION FOR 10- AND 20-FOOT MODELS.

Beam-draft ratios higher than 2.92 have greater pressure resistance per ton of displacement, lower ratios less because R_p varies with the bow angle.

The procedure to obtain R_p/W for any value of B/D is as follows: suppose $B/D = 2.40$, and $\dfrac{W}{(0.01L)^3} = 200$. The original curves give,

$$\begin{array}{lll}
R_p/W = 1.70 \text{ for } B/D = 3.75, & B/D = 2.40 \\
R_p/W = \underline{1.23} \text{ for } B/D = \underline{2.25} & B/D = \underline{2.25} \\
\text{Difference} \quad = 0.47 \qquad\qquad 1.50 & \qquad\quad 0.15
\end{array}$$

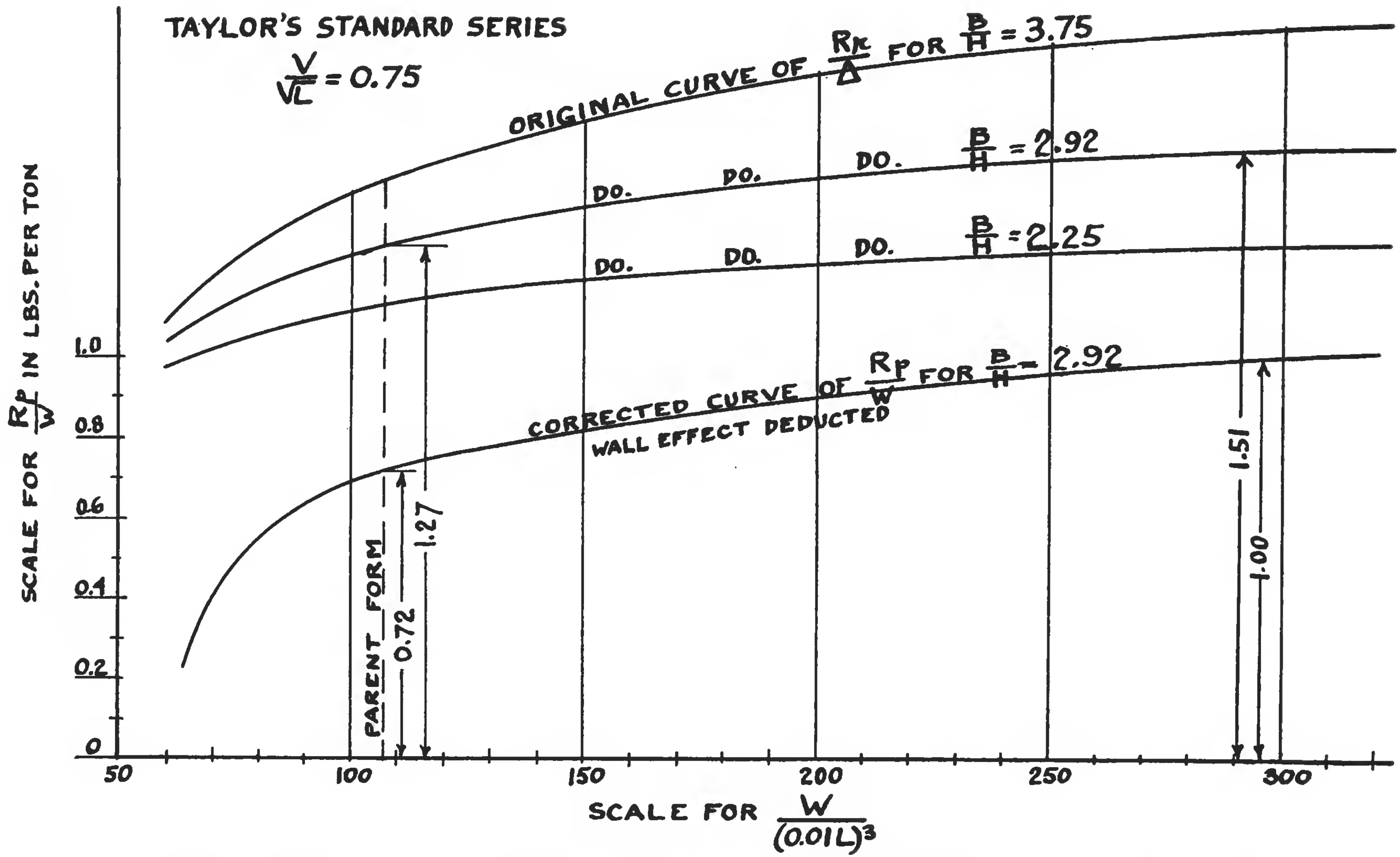


FIG. 25. CORRECTED CURVES FOR PRESSURE RESISTANCE—TAYLOR'S STANDARD SERIES, AT $V/\sqrt{L} = 0.75$.

$R_p/W = 1.44$ for $B/D = 2.92$ (original), corrected $R_p/W = 0.90$ lb./ton.

It can be proved that the difference in B/D is directly proportional to the increase in R_p/W above its lowest value. Hence, 0.15 divided by $1.50 = 0.10$ which, multiplied by $0.47 = 0.047$; $1.230 + 0.047 = 1.277$, and this amounts to 88.2% of the original value 1.44 of R_p/W for $B/D = 2.92$. Finally, the corrected value 0.90 times $0.882 = 0.794$ pounds per ton which is the answer sought, quite a bit lower than the original value 1.277. For vessels in salt water, multiply by 1.025.

If the B/D ratios are above 3.75 or below 2.25, the answer is obtained in the same manner but more easily from a diagram based on B/D, with ordinates of the original values of R_p/W for the displacement-length ratio as given. For instance, in the case just treated, the end ordinates would be 1.70 and 1.23, and a straight line from one to the other shows all intermediate values of R_p/W that can be corrected as indicated above. By extension of the straight line, any value is obtained directly.

The procedure here outlined is not mathematically correct, but founded on the assumption that the pressure resistance R_p varies in the same manner whether the bow angle increases with the displacement-length ratio as in the Taylor Standard Series, or is constant for each $V/\sqrt{L}$, as in the writer's hull form of maximum efficiency and earning capacity. Careful estimates have proved that this assumption is very nearly true, and in any case the nearest one can get to actual values, considering the infinite varieties in hull form and in curve of areas.

Pressure Resistance. Froude and other pioneers found that after deducting the frictional resistance from the total model resistance, there remained another kind of resistance that they termed *residuary*. It included resistance from the divergent bow waves, from the transverse waves, from eddies and whatnot. But since all these parts were caused by pressure differences around the model, a far simpler name for them is *pressure resistance* in contrast to frictional resistance, in which pressure differences do not act at all.

The pressure resistance R_p is originated at the bow of the vessel, both as regards the divergent and the transverse wave systems. If a wedge-shaped body is pushed through water, apex foremost, a wave of certain length is created on each side of the wedge opposing the

movement, thus increasing the resistance. Simultaneously another wave is formed, its crest running transversely to the ship. No explanation has been given how the transverse wave system is formed but, from the experiments of Froude and others, it is evident that its waves can be seen best along the middle body of the ship, and that the crests are exactly opposite those of the divergent lines. The latter fact seems to indicate that the transverse waves are formed by the continuation of the divergent waves toward the sides of the ship. But why should the particles of one wave system revolve in circles parallel to the course of the ship, and of the other in circles at an angle to the same course, if the trochoidal wave theory is correct?—And how is it possible for the circles to change their planes of revolution in the twinkling of an eye?—Whatever the answer, the ship's wave system is complete proof that the streamline pressure and its resulting waves are inactive on a surface vessel, and that wave pressures take their place. The streamline theory would have a wave crest at each end of the ship, and a wave hollow in the middle. Further, the net resistance of these waves would be nothing. Our experience tells us that just the opposite is valid in ships, that there are many waves along the sides of the vessel, and that these waves cause a big resistance, the bigger, the higher the speed of the ship.

As long as the speed is uniform, the waves follow the ship exactly, and are, in fact, stationary in relation to the ship, yet are left behind from the start. The very simple explanation is that it takes the waves the same time to "*advance backwards*" from the crest to the hollow, and from the hollow to the next crest, and that in this time the ship is advancing one wave length. The fact of the waves being stationary means that their pressures against the ship are stationary, too. But since the wave length increases with the speed, the pressures change their positions on the ship's sides, which change has a tremendous effect on the resistance, especially at high speed-length ratios, necessitating the choice of favorable $V/\sqrt{L}$ as already stated.

When the water is pushed aside by the bow, and when it closes in behind the stern, waves are created that rise to a certain height very near to the hydraulic velocity head $v^2/2g$, where v is measured in feet per second, and $g =$ (32.16 feet per second)^2. The velocity is the divergent speed $V \times \sin a$, with V in knots, $a =$ the angle of incidence at the bow in degrees. Most old theories made the head $= \dfrac{V^2 \times \sin^2 a}{2g}$,

but photographs and measurements all agree that the head is proportional to sin a, not to $\sin^2 a$. Expressed in knots and feet,

$$h = \frac{V^2 \times \sin a}{11.50} \tag{43}$$

It takes some time for the water to rise to h, and during that time the ship is advancing a certain length; but the phenomenon is very intricate, as friction against the sides of the vessel and other factors shorten the length to $\frac{V^2 \times \sin a}{5.75}$, exactly twice the height h in formula (43). For estimating purposes the projection of this length is needed, or $l \cos a$, hence

$$l = \frac{V^2 \times \sin a \times \cos a}{5.75} \tag{44}$$

The correctness of these formulas can be tested by photographs of ships in smooth water, and is confirmed by experimental data. Since the length of a wave $l_w = V^2/1.80$ as stated, it is clear that h, l, and l_w bear a close relation to each other, dependent only on the angle of incidence a.

As regards the stern waves, all their dimensions are shortened by the wake aft of the ship, and the so-called wave-making length is less than the ship's *LWL*, contrary to the accepted Froudeian theory, with its many errors.

Another error lies in the assumption that the speed of the water along the ship's surface is influenced by streamline action, or worse still, by the obliquity of the surface. Since no forces are acting either for acceleration or retardation, the speed over the surface must be constant $= V$, as it is assumed in estimates of frictional resistance. This error reaches its profoundest depth in the *Rankine Augmented Surface*, now considered obsolete.

The normal, or square to the surface, pressure on a wedge or a plane has been investigated by many scientists since Joessel, who obtained resistance in pounds $= 4.62AV^2$. Recent investigations by the writer made the constant $= 4.58$, a very slight difference. For the bow wave resistance, rudder experiments are more conclusive, resulting in a formula $N = CSV^2 \times \sin a$, where

N = normal pressure on a plane, in pounds

C = a constant, slightly dependent of shape of plane
S = area of plane in square feet
V = speed of ship in knots
a = angle of incidence at the bow
Now $S \times \sin a$ is the base area of a wedge or cone $= A$, hence

$$R_p = CAV^2 \tag{45}$$

which formula expresses the relation between resistance, speed, and effective area. In a ship A means the sectional area at the initial crest of the bow waves as stated in formula (44). Over this area at the bow, or $A/\sin a$, the water particles are accelerated and thereby cause the resistance R_p. Aft of the initial crest the waves are stationary, and crest pressure balanced by trough pressure except when wave interference occurs, of which more presently.

If the wave crests should happen to stand symmetrically on the profile of the ship, and its both ends were alike, there could be no wave resistance except from the pressure on the bow by the initial crest. However, a ship's ends are never alike, and the bow waves decrease in height and pressure aft of the initial crest, which facts modify the conditions for balance of the wave systems. Professor H. S. Sadler, of the University of Michigan, many years ago made some important experiments on the relative lengths of forebody and afterbody in 10-ft. models, proving that the angles of incidence at the stern influenced the wave resistance nearly as much as that of the bow, and that both angles must be included in any formula for pressure resistance. The design of his parent model has been lost, but the curve of areas is shown in reference 37. Both ends of this curve were alike, which increases the usefulness of the experiments.

When both ends are thus alike, it can be proved geometrically that the sum of the bow and stern angles, and the pressure resistance, is minimum when both angles are alike. Sadler found that the minimum was reached when the length of the forebody was 54% of the model's LWL. The difference is accounted for by the fact that while the areas of the end sections were alike, the shapes of the sections were different as usual in hull forms. The effect of the stern angle can be equalized by taking 98% of the actual value for insertion in the pressure resistance formula.

To add to our difficulties, the form of the ship ends are seldom flat in the sections, and the angles of incidence differ in value from the

LWL to the keel. These angles are measured in the body plan square to the sections, not horizontally or vertically. In our difficulties we are helped by the fact that the trochoidal wave pressures are highest near the surface of the water, and decrease quickly downwards. Hence we need not go farther down than 25% of the height of the bow waves, formula (43). Pressures below that level are not considered, as their net effect on resistance is practically 0.

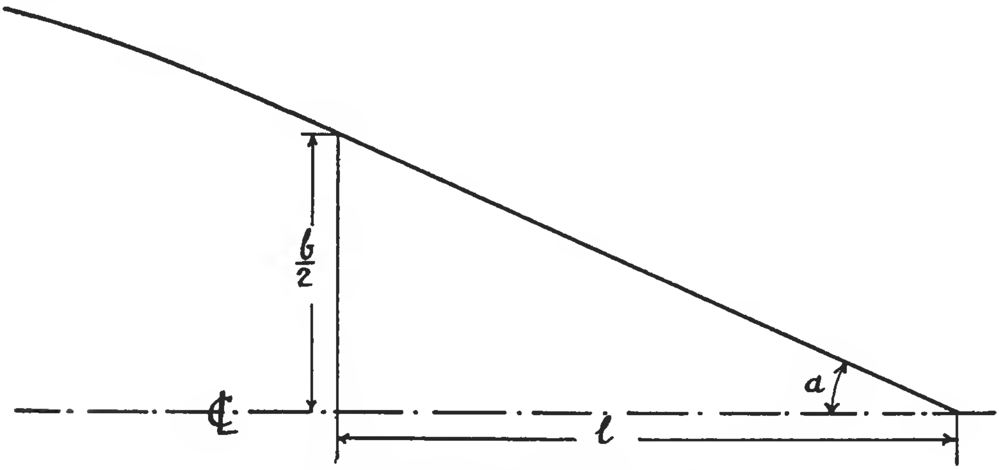


FIG. 26. BOW WAVE PARTICULARS.

Consider now a longitudinal section along a diagonal drawn square to the surface of the hull at the bow, Fig. 26. The first step is to obtain the area A in formula (45), at the distance l from the end of the water line at the bow, formula (44). To begin with, A includes the entire transverse sectional area from *LWL* to keel, and equals Cbd, where C is the section coefficient, b the width of the section, and d the depth. From the figure is seen that $b/2 = l \tan a$ which equals

$$\frac{V^2 \times \sin a \times \cos a \times \tan a}{5.75}$$

according to formula (44). The $\tan a = \sin a/\cos a$, and $b/2 = V^2 \times \sin^2 a \times$ constant.

This value of A inserted in formula (45) gives the pressure resistance at the bow,

$$\begin{aligned} R_p &= d \times \sin^2 a \times V^2 \times V^2 \times \sin a \times \text{constant} \\ &= d \times \sin^3 a \times V^4 \times \text{constant} \qquad (46) \end{aligned}$$

The formula is developed on the assumption that the water line is straight between the section at $b/2$ and the stem. If not so, or if the

curve of areas is hollow, $b/2$ is no longer equal to tan a but worse than that, the exponent of V is greater than 4. For instance, if the curve of areas were a concave parabola of the second order near the stem, the exponent of V would be 6 instead of 4. It means that, although the pressure resistance might be very low at first, as the speed increases it would rise much faster than if the curve of areas were straight. Suppose the speed is doubled: $2^4 = 16$, but $2^6 = 64$, an increase of 400%.

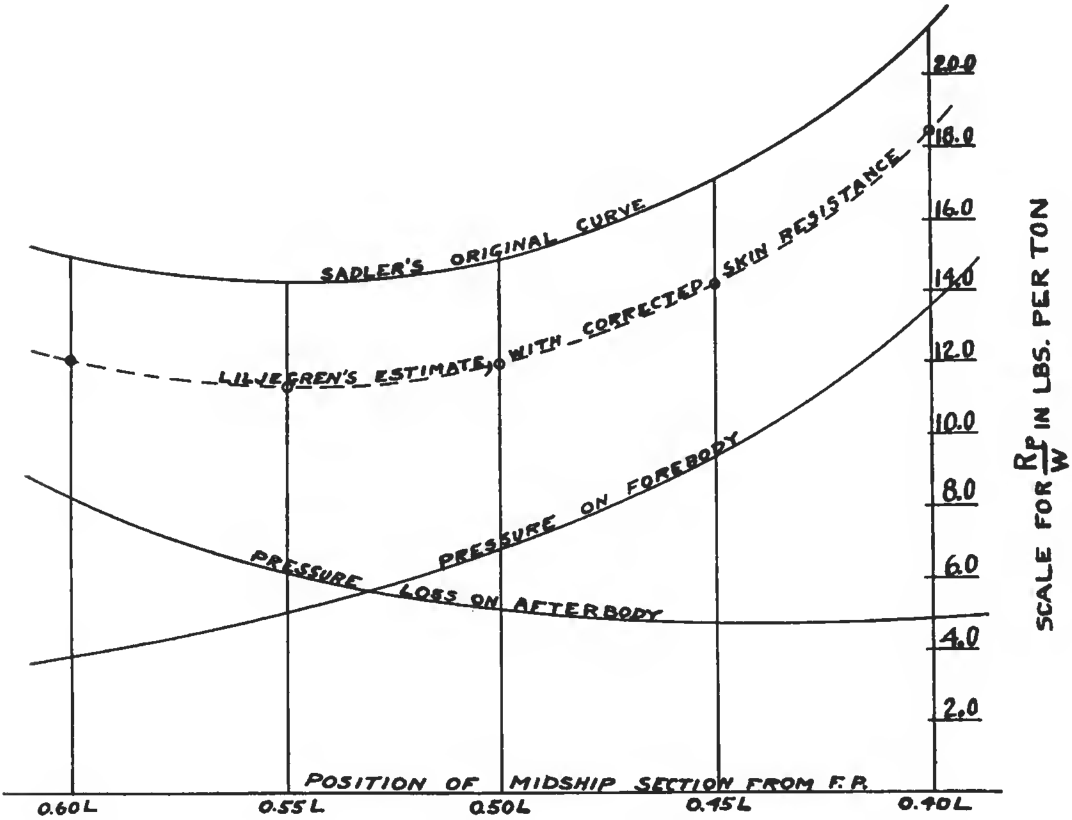


FIG. 27. SADLER'S EXPERIMENT.

The reasoning of the pressure resistance applies just as well to the stern waves according to Professor Sadler's experiments, the pressure curves of which are shown in Fig. 27, corrected for frictional resistance but not for wall effect. The speed-length ratio was 1.20, high indeed, but the models were very well-shaped, long and narrow. The midship section area and the speed being constant, wall effect could not materially alter the result of the experiments.

The curve of pressure resistance derived from formula (46) is a parabola of the fourth order, and represents the mean resistance not

affected by wave interference. It should also be noted that while the Sadler experiments gave least resistance when the forebody was longer than the afterbody, the case is quite different for a screw-propelled

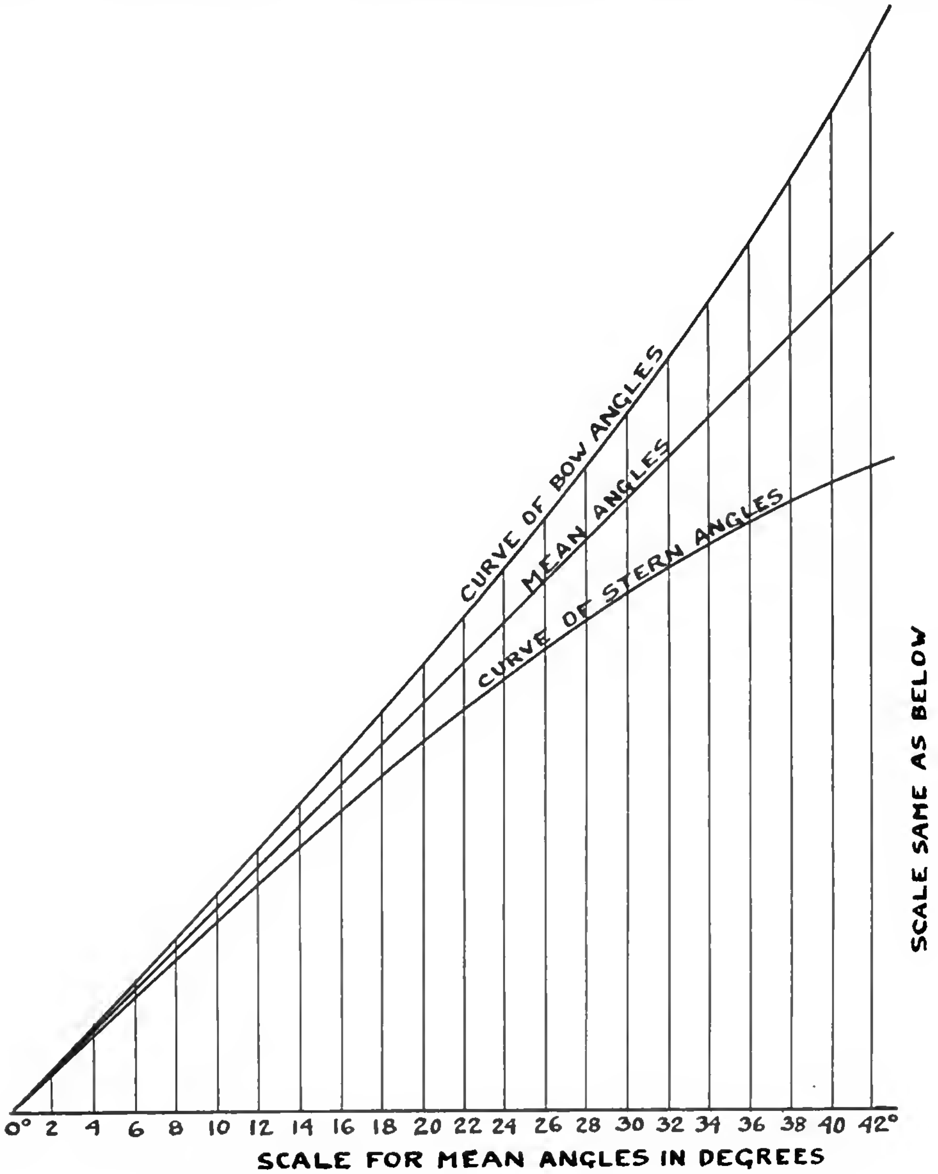


FIG. 28. CURVES OF BOW AND STERN ANGLES.

vessel, the stern angle of which should always be less than the bow angle in order to obtain an easy flow to the propeller, and a high efficiency. Fig. 28 indicates the best ratios of bow and stern angles for given mean angles. The mean of the bow angle plus 98% of the stern angle should be used in formula (46).

It is not the absolute resistance, but the resistance per ton of displacement that must be a minimum, as repeatedly stated.

1) $\frac{R_p}{W} = K_2 \frac{d \times \sin^3 a \times V^4}{C_b \times LBD} = K_2 \frac{\sin^3 a \times V^4}{C_b \times LB}$, assuming that as usual $d = D$. Draft is not influencing R_p/W.

2) $\frac{R_f}{W} = K_1 \times \frac{WSV^2}{W} = K_1C \frac{(c_0B + 2c_1D)LV^2}{C_b \times LBD}$. *See* formula (37).

Make $D = c_2B$, $\frac{R_f}{W} = K_1C \frac{(c_0 + 2c_1c_2)BLV^2}{C_b \times LBD} = K_1C \frac{(c_0 + 2c_1c_2)V^2}{C_b \times D}$

The bow sections are almost vertical near the *LWL*, and Fig. 26 can also represent the *LWL* plane. If extended to the length of the forebody l_f, $\tan a = \frac{c_3B}{2l_f}$, and $B = \frac{2l_f \times \tan a}{c_3}$. If c_2B is substituted for D in the denominator derived in the main formula,

$$\frac{R_f}{W} = K_1C \frac{(c_0 + 2c_1c_2)}{c_2} \times \frac{c_3}{C_b \times 2l_f \times \tan a} \times V^2$$

K_1, K_2, C, c_3 are constants, the values of which are determined from the design of the ship. In the writer's parent design with the *LWL*-curve ending in a straight line forward, $c_3 = 1.57$, but is greater for a hollow *LWL*-curve, at the shoulder.

The term $\frac{c_0 + 2c_1c_2}{c_2} = 4c_1$ when $(c_0B = 2c_1D)$, the condition of minimum wetted surface, *see* formula (38), which condition must be fulfilled when the minimum resistance is sought. This fact is very important, because the skin resistance usually amounts to 80% of the vessel's total resistance.

PROOF.

$$\text{The term} = \frac{c_0 + 2c_1c_2}{c_2} = \frac{c_0 + 2c_1 \times D/B}{D/B} = \frac{c_0B + 2c_1D}{D},$$

$$\text{which equals } \frac{4c_1D}{D} = 4c_1$$

The formulas derived under (1) and (2) above, added together, equal the total resistance per ton of the ship's displacement. In order to

make it a ***minimum***, differential calculus must be introduced, and first derivatives of (1) and (2) added and equal to 0. As there are many variables in both formulas, only partial derivatives can be obtained, and the question is, which variable should be preferred. At sight, the beam B seems best, but when the derivatives are solved, B is eliminated and no result comes forth. By deciding on the mean bow angle a, a solution in terms of a is possible. By selecting a, all other variables become constant according to the rules of derivatives.

In the first formula for R_p, the differential of $\sin^3 a = 3 \sin^2 a \times \cos a \times \delta a$ or, in symbols of calculus, $\delta(\sin^3 a) = 3 \sin^2 a \times \cos a \times \delta a$. In the second formula $\delta\left(\frac{1}{\tan a}\right) = -\frac{1}{\sin^2 a}$, a negative term that is needed in the derivative formula because otherwise either R_f or R_p would be negative, which is impossible. The fundamental formula is

$$\frac{\delta(R/W)}{\delta(a)} = \frac{\delta(R_f/W)}{\delta(a)} + \frac{\delta(R_p/W)}{\delta(a)} = 0 \qquad (47)$$

which interpreted means that the equations of the resistances per ton of displacement are differentiated in respect to a, the mean of the bow and the stern angles, as explained on page 123.

Part III-B

Naval Architecture As An Exact Science

DESTROYER U.S.S. *Anderson*. SPEED-LENGTH RATIO, 1.98. *Official U. S. Navy photograph.*

PART III-A and guesswork in ship designing are now left behind, and our attention turned to *exact science* for the first time in shipbuilding history. There are not many exact sciences left today, the principal ones being mathematics, geometry, and in a lesser sense, astronomy. The remainder change every few years, and what is undying truth today, is a perennial laughingstock tomorrow. Still, pioneering is necessary, and without the first few faltering steps no science would be possible.

HULL OF MINIMUM RESISTANCE

The form of minimum resistance has been sought in many ways, the latest through model experiments. It does not seem to have dawned upon the experimenters that the laws of frictional resistance are quite different as applied to models and to ships, and that this well-known fact *a priori* demands quite different dimensions of the model and the ship. Further, full-size vessels over 100 ft. long are not affected by excess resistance, nor by laminar flow, longitudinal edge effect, and wall effect, as explained in preceding Parts. It is thus quite useless to expect a hull form of least resistance from any model experiments, however carefully made. But any problem involving least or most can be solved by calculus of maxima and minima, a special division of differential calculus. A prerequisite for solution is that the fundamental equations defining the variables can be unfolded into first and second derivatives. The two fundamental equations in this case are the formulas for frictional resistance and for pressure resistance, and the result of the differentiation is shown in formula (47), page 125 in Part III-A.

A great many formulas have been tested in this manner, and results compared with actual best practice until the present simple ones, formulas (34) and (46), were obtained. (*See* pages 97 and 121 respectively.)

Returning to formula (45), and inserting the constant factors,

$$\frac{\delta(R_p/W)}{\delta(a)} = K_2 \frac{3 \sin^2 a \times \cos a}{C_b \times LB} \times V^4,$$

and $$\frac{\delta(R_f/W)}{\delta(a)} = -K_1 C \frac{(c_0 + 2c_1c_2)}{c_2} \times \frac{c_3}{C_b \times 2l_f \times \sin^2 a} \times V^2$$

In the upper equation there are no reductions to be made, but in the second $\dfrac{c_3}{2l_f} = \dfrac{\tan a}{B}$, and $\dfrac{(c_0 + 2c_1c_2)}{c_2} = 4c_1$, as previously explained for ships with a minimum wetted surface. It will be found later that this condition cannot be fulfilled in the case of high-speed vessels, for which the factor in question must be estimated separately. Adding the two equations and putting them equal to 0 according to formula (47) means that both are equal. Hence

$$K_2 \frac{3 \sin^2 a \times \cos a}{C_b \times L^3} \times V^4 = K_1 C \times 4c_1 \times \frac{\tan a}{C_b B \times \sin^2 a} \times V^2$$

$$\text{Tan } a = \sin a / \cos a,$$ and

multiplying both sides and reducing, results in

$$\sin^3 a \times \cos^2 a = \frac{K_1}{3K_2} \times C \times 4c_1 \times \frac{L}{V^2} \tag{48}$$

Formula (48) is the exact mathematical solution of formula (47). The angle of incidence is the mean of the bow angle and the stern angle measured as previously explained, C is the wetted surface constant, $4c_1 = 4.0$ for the old-type ship, and about $= 3.8$ for a cruiser stern, K_1 and K_2 are constants depending on the form of the hull, and on the roughness of the wetted surface, respectively.

All the constants are known for a ship in trial trip conditions—that is, first rate—but these stay on only a few weeks. Besides, head seas on some routes may necessitate a smaller angle than formula (48) indicates, and smooth water routes may permit larger angles. It follows that here as in every other physical research, dependence must be made on experiments and experience. Some vessels were built with far too fine hulls, others with hulls having more earning capacity; but the writer knows of only two types that possessed great earning capacity from the keel up: the *North East Coast colliers* in Great Britain, whose negative propeller ship, however, proved that their sterns were far too full, requiring enormous engines, and the *American Great Lakes vessels*, still going strong, and considered to be the most economical cargo ships in the world. Until recently, their speed or, rather, speed-length ratio, was very low, under 0.50.

In designing a merchant ship for highest investment value, or a warship for highest efficiency value, it is of course necessary to get a

hull of least resistance per ton of displacement, and to choose the most favorable speed-length ratio. Inspection of formula (48) shows that the bow angle is proportional to the square of the inverted speed-length ratio, or to $(\sqrt{L}/V)^2$. But the actual amount of the pressure resistance varies with the interference of the bow and the stern wave series, and it becomes necessary to choose a $V/\sqrt{L}$ where there is no interference. After many attempts, the most suitable $V/\sqrt{L}$ was found to equal 1.0, which value incidentally simplifies formula (48) still further.

After investigating thousands of successful ship designs, the writer finally arrived at the conclusion that the safest bow angle at $V/\sqrt{L} = 1.0$ was 15°, but that the angle varied a little with the fullness of the midship section. Now $\sin^3 15° \times \cos^2 15° = 0.01635$, and formula (48) reduces still further to

$$\sin^3 a \times \cos^2 a = 0.01635 \frac{L}{V^2} \tag{49}$$

from which formula all angles of incidence have been calculated.

At all favorable speed-length ratios, these angles are increased, since the pressure resistance is decreased while the frictional resistance per ton of displacement remains unaltered. These favorable $V/\sqrt{L}$ have been stated to be 0.63, 0.72, 0.85, 1.11, to which can be added 2.04 as the highest where interference is favorable. Speed-length ratios such as 0.81, 0.90, and 1.02 are neutral and located on the mean resistance curve. All $V/\sqrt{L}$ vary slightly with the bow angle because the distance to the first crest of the bow waves is proportional to $\sin a \times \cos a$, the maximum of which is at 45°.

Formula (49) proves that within the same $V/\sqrt{L}$ the mean angle of incidence remains constant, for if the speed-length ratio is constant, the right side of the equation is constant too, and thus the left side, as a consequence.

THIS CONCLUSION STATES THAT PARENT DESIGNS WITH INCREASING BOW ANGLES CAN NEVER REACH MAXIMUM EARNING CAPACITY.

HOW TO DESIGN A PROFITABLE CARGO SHIP

Suppose either the speed or the length on the water line has been determined from trade and route conditions, also $V/\sqrt{L} = 0.72$. With each V or L goes a certain displacement, and if not large enough for the cargo capacity needed, a lower $V/\sqrt{L}$ or a higher V must be used.

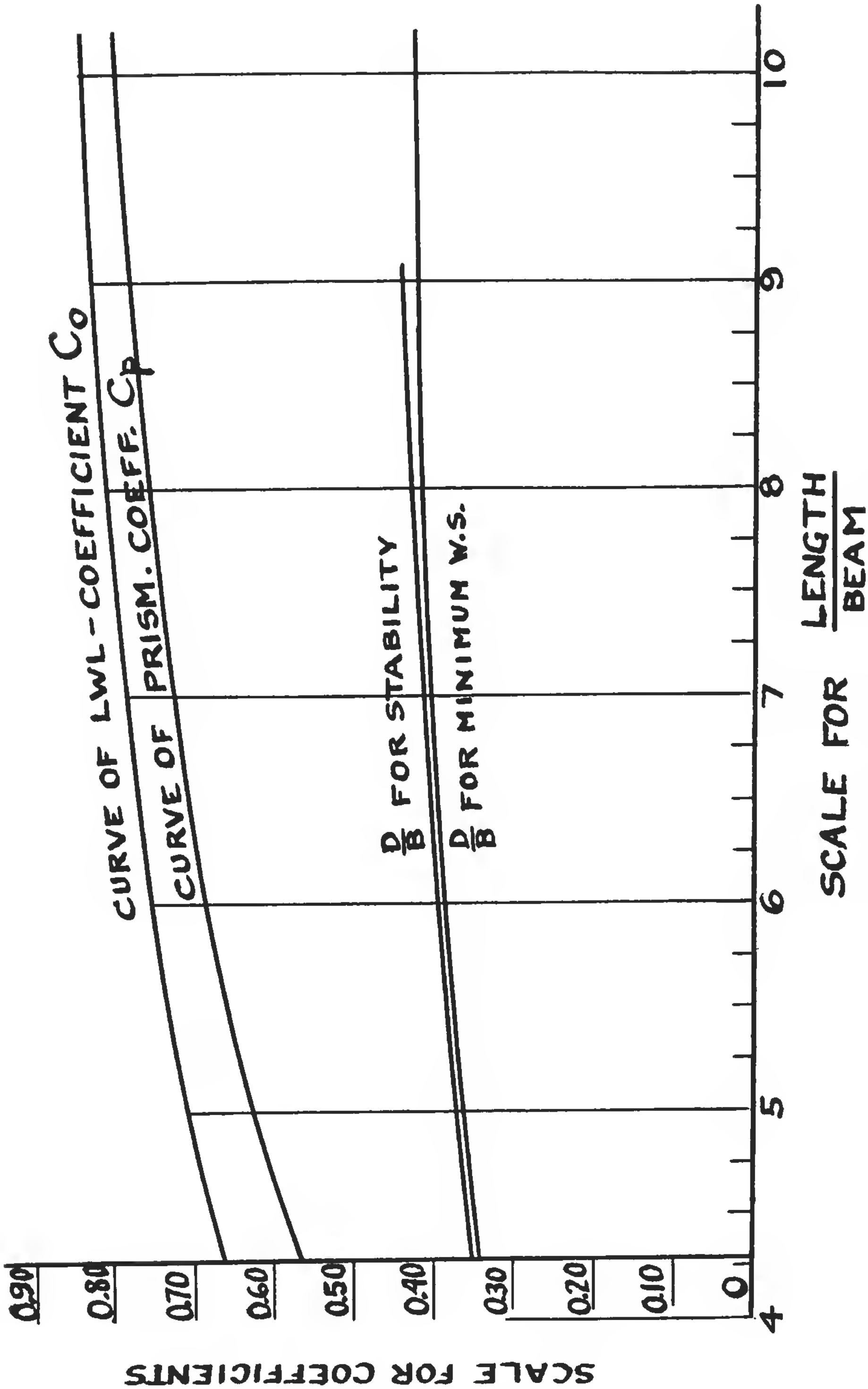


FIG. 29. CURVES OF HULL FORM COEFFICIENT.

In any case the mean bow angle is fixed, and thus $B = \dfrac{2l_f}{\tan a}$; but in this equation l_f (the length of the forebody) is still unknown, hence another equation must be added. The most suitable is formula (8), page 33, by which B is estimated but must be checked for stability reasons by formula (18). B being determined, D is also obtained from formula (17), page 66.

Turn now to Fig. 29, and from the proper L/B-ratio get the values of C_0, C_1, C_p, and C_{ws}. C_x, the midsection coefficient, is assumed to equal 0.985, from which C_b is estimated, and finally the displacement.

There is yet to determine l_a (the length of the afterbody). As soon as the bow angle a is known (22° 48′ at $V/\sqrt{L} = 0.72$), the stern angle b is picked off from Fig. 28 as 19°. Over LWL drawn to any scale, set two triangles having angles of 22° 48′ at stem, and 19° at stern. If both ends were alike in the design, $l_a = l_f$, but in this design,

$$\frac{l_a}{l_f} = \frac{\tan a}{\tan b} \times 0.865 \sqrt{1 + \frac{4D^2}{B^2}}$$

The coefficient 0.865 before the radical sign is the part that the height of the after trajectory is shorter than the bilge diagonal, as represented by the term under the radical. l_f being already estimated, l_a is now obtained, and $l_m = L - l_f - l_a$, completes the estimates of the principal dimensions, after which proceed as outlined in preceding Parts, page 36 *ff*.

A word now about the manner in which the *hull form coefficients* in Fig. 29 have been computed.

1) *Prismatic coefficient* C_p.—In the parent design without any parallel middle body, $C_p = 0.566$, and the same value applies to the forebody and to the afterbody of all derived designs, hence

$$C_p = 0.566 \times \frac{l_f + l_a}{L} + \frac{l_m}{L}$$

In the parent design, $L/B = 4.25$, and suppose we want to find the C_p for $L/B = 5.0$. Then, $5.0 - 4.25 = 0.75$ which divided by $5.0 = l_m/L = 0.15$. Deduct l_m/L from $1.00 = 0.85 = \dfrac{l_f + l_a}{L}$ which multiplied by $0.566 = 0.481$. Finally, $0.481 + 0.150 = 0.631$, the required C_p.

2) *LWL-coefficient* C_0.—In the parent design, $C_0 = 0.662$ for the ends. In the same manner, if L/B is assumed $= 5.0$, $C_0 = 0.713$.

3) The remaining hull form coefficient is obtained in the same manner as can be applied to any speed-length ratio, except that the parent L/B varies somewhat with $V/\sqrt{L}$ as follows:

$\frac{V}{\sqrt{L}}$	0.63	0.72	0.85	1.11	2.04
$\frac{L}{B}$	3.92	4.25	4.85	5.38	6.07
Bow Angle	25°	22° 48′	20° 28′	17° 54′	15° 54′
Stern Angle	20° 20′	19° 0′	17° 18′	15° 33′	14°

Trajectory Curve. The best way to shape the hull longitudinally is by trajectories that are run square to the transverse sections in the body plan, which means square to the surface of the ship or very nearly so. The Laws of Mechanics tell us that acceleration or retardation need a causative force, hence there results a loss of power, a resistance. In order to make this loss a minimum, the path of the force must end in a straight line, because only a curve can create acceleration across it. Since a ship must possess beam, the trajectories must consist of curves, and the natural line that fulfills both conditions is termed an *elastic line*—the curve a bent elastic plank takes. Unfortunately, the elastic line is very hard to lay out on paper, and its place is taken with advantage by the *sinus line*, whose equation is $y = \sin x$.

In laying out this curve, either l_f or l_a is made the base, divided into 90°, and the height is made = 1.0, since $\sin 90° = 1.0$. In the same manner, at 1/3 of the base, or 30°, the ordinate would be 0.50, the sine of 30°. The height of the curve, of course, is the width of the trajectory measured to the same scale as the body plan in the design. Generally, it is enough to erect ordinates at every 15°, since the curve is very fair. The angle between the curve and the base is equal to the bow or the stern angle as the case may be. The position of the transverse sections is marked on the curve, from which the widths are lifted directly to the body and to the half-breadth plans.

The sinus line is very useful as a curve of areas when the ship's keel is a straight line, otherwise it is a double-curved or S-line that makes the best hull form. In any case the curve extends only over the ends of the ship, not over the middle body.

BOW WAVE INTERFERENCE

Waves possess a certain energy, half kinetic and half potential, that expresses itself as resistance to the vessel's motion. At the very bow, there is a dynamic pressure as far aft as to the crest of the wave, but everywhere else the pressure is wholly static as long as the speed of the ship remains constant. For every crest there is a trough or hollow, and two intermediate points causing over-pressure, under-pressure, and average pressure. The disposition of the crests and troughs over the surface of the vessel determines if there is a net increase, or a net decrease in pressure resistance. At a certain speed the waves are placed symmetrically about the midship section in the profile, and their net effect is very small—in fact would reach zero if both ends of the ship were alike. In this case the wave system would be in nearly perfect balance, the resistance made up entirely by friction and dynamic pressure at the bow.

But both ends are never similar, and the bow waves decrease in height with each crest, hence their pressure is both dynamic and static. At the intermediate points one meets average resistance, the ordinates of which vary as the fourth power of the speed if the latter is laid off as abscissae. The length of the waves increases as the square of the speed; when the procession of waves travel over the ship from stem to stern, a series of humps and hollows are formed in the resistance curve. If, on the other hand, the speed be constant but the ship's length varying, the humps and hollows are just as much in evidence.

This phenomenon is termed wave interference. It was first discovered by W. Froude, who in 1877[32] made experiments for the British Admiralty, but his explanation of the facts was entirely erroneous. Froude and all his non-critical followers imagined the stern waves reached back, or rather forward, and caused the undulations in the resistance curve in connection with the bow system. But he never tried to explain how the stern waves could possess such a large effective force. Naturally, the stern waves, having no direct connection to the ship, and situated wholly behind the ship, cannot alter its resistance like, for instance, the screw propeller.

The stern wave is caused by the concussion of the water particles closing in around the stern, just as the bow wave is created by the dividing shock at the bow.

The writer in 1898 developed a theory of the interference based on the Froude experiments, the only ones available at the time. The theory has since been corroborated by later model experiments, notably by Taylor in the Washington basin, but all experiments suffered much from wall effect at high speeds. The mean resistance line, instead of staying horizontal at constant speed, curved upwards for the shortest models. This was more in evidence in the Froude experiments even when corrected for skin resistance.

Starting with the symmetrical arrangement of the wave crests, and calling that the minimum interference, the writer succeeded in evolving for the speed-length ratios at which a minimum, a maximum, or a neutral interference should occur, on the assumption that both ends of the ship were alike. Between the maximum and the minimum interposes one-half wave length, and the distance from the stem to the first crest, and from the last crest to the stern, were supposed equal. The ship length L was made up by $2l$, formula (44), and nl_w, n being the number of waves on L. Or

$$L = 2l + nl_w$$

$$2l = 2 \times \frac{V^2 \times \sin a \times \cos a}{5.75}, \frac{1}{5.75} = 0.174, \frac{1}{1.80} = 0.555,$$

$$l_w = \frac{V^2}{1.80}, \text{ and the equation}$$

$$L = V^2 (\sin a \times \cos a \times 2 \times 0.174 + 0.555n). \ \frac{L}{V^2} \text{ is the inverted}$$

speed-length ratio squared, hence

$$\text{Minimum at } \frac{V}{\sqrt{L}} = \sqrt{\frac{1.0}{0.348 \sin a \times \cos a + 0.555n}} \tag{50}$$

Maximum occurs, as mentioned, one-half wave length farther away, which means that the term to the right of the plus sign should read $0.555(n + 0.50)$, otherwise the equation remains unchanged.

The result of formula (50) compared to Froude's experiment is shown by Fig. 30, and while the agreement is good for $n = 1.0$ and $n = 2.0$, later experiments have distinctly proved that the distance from the last crest to the stern is greater than l, and the combined distances greater than $2l$, sometimes as much as $2.50l$. This could be expected, since the stern is usually finer in form than the bow. It should be noted

that the bow angle a influences the $V/\sqrt{L}$ at which a minimum interference should occur.

In Fig. 30 is seen that the actual interference is less than the theoretical because in the latter no account has been taken of the action of internal friction in the wave body, nor of the friction against the sides of the ship. This friction, even if small, naturally reduces the effect of the interference.

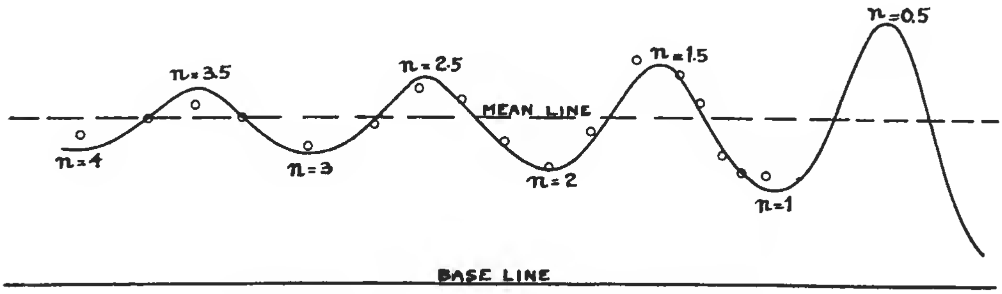


FIG. 30. CURVE OF INTERFERENCE (FROUDE).

Interference Effect. The effect of the wave interference is measured by the percentage increase of pressure resistance at a maximum, and by the percentage decrease at a minimum. In order to find the correct resistance, these percentages must be added or deducted from the mean resistance. As the percentages actually measure the *amplitude* or the amount of deviation from the mean resistance, the term amplitude is best.

The energy of the wave system created by the ship is divided into two parts and apportioned between the divergent waves and the transverse waves. The former become entirely separated from the ship, and only the energy of the latter can produce the undulations in the resistance curve. The mean resistance and thus the mean speed being constant, energy is proportional to resistance, and vice versa.

Let E = energy of the transverse waves, per foot of width
e = energy of the divergent waves, per foot of width
l_t = length of the transverse waves
l_d = length of the divergent waves

Since the length is proportional to the square of the speed,

$$\frac{l_d}{l_t} = \frac{V^2 \times \sin^2 a}{V^2} = \sin^2 a$$

DESTROYER U.S.S. *Flusser*, SHOWING BOW WAVES NEARLY HEAD ON. SPEED-LENGTH RATIO, 1.98. *Official U. S. Navy photograph.*

The energy of a wave of trochoidal form $= 8lh^2$, and although their lengths are different, the divergent and the transverse waves have the same height. $e/E = \sin^2 a$ in consequence. The divergent waves make an angle of $(90° - a°)$ with the transverse waves. To get the equivalent width of the divergent waves, we must divide e/E by $\sin a$. Thus

$e = E \sin a$, and the total energy $= e + E = E(1 + \sin a)$.

Hence $E = \dfrac{\text{Total Energy of the Wave Systems}}{1 + \sin a}$ at the first crest. For every crest aft of the first, is dispersed over a wider area, and the energy per foot reduced as 1/2, 1/3, 1/4, etc., or as

$$E_n = E \times \frac{1}{n+1} \text{ generally.}$$

$$\text{Hence } E_n = \frac{\text{total energy of the wave systems}}{(1 + \sin a)(n + 1)} \tag{51}$$

Energy can also be likened to horsepower, and is proportional Rl or to resistance times length. At any particular speed, length of the wave is constant too, and $E = CR$. Instead of E in formula (51), R can be inserted without altering the equation that now becomes an expression for the force that causes the undulations in the resistance curve. If we make

R_p = actual pressure resistance as shown in model tests
R = total mean resistance
n = number of wave lengths in the length of the ship

Minimum:
$$R_p = R - \frac{R}{(1 + \sin a)(n + 1)}$$
$$= R\left(1 - \frac{1}{(1 + \sin a)(n + 1)}\right) \tag{52}$$

Maximum:
$$R_p = R\left(1 + \frac{1}{(1 + \sin a)(n + 0.5)}\right) \tag{53}$$

At all minima there are hollows in the resistance curve, and at all maxima there are humps in the curve. After the last hollow there is no more interference. In formulas (52) and (53) the amplitudes of the humps and the hollows are shown by the term to the right of the minus sign and of the plus sign. For instance, suppose $a = 20°$, and

$n = 1.0$, the amplitude from formula (52) is $\dfrac{1}{1.342 \times 2.0} = 0.372$ which means that over 37% of the pressure resistance would be gained by the ship at the corresponding $V/\sqrt{L} = 1.225$, according to formula (50). A similar percentage would be lost to the ship if a maximum had been chosen. It is significant that modern vessels do not show any undulations in the resistance curves, being too lean in the body, thus possessing little carrying capacity per ton of displacement. Exceptions were the U. S. Shipping Board models tested by the Washington Model Basin—models that showed exceptionally large humps and hollows.

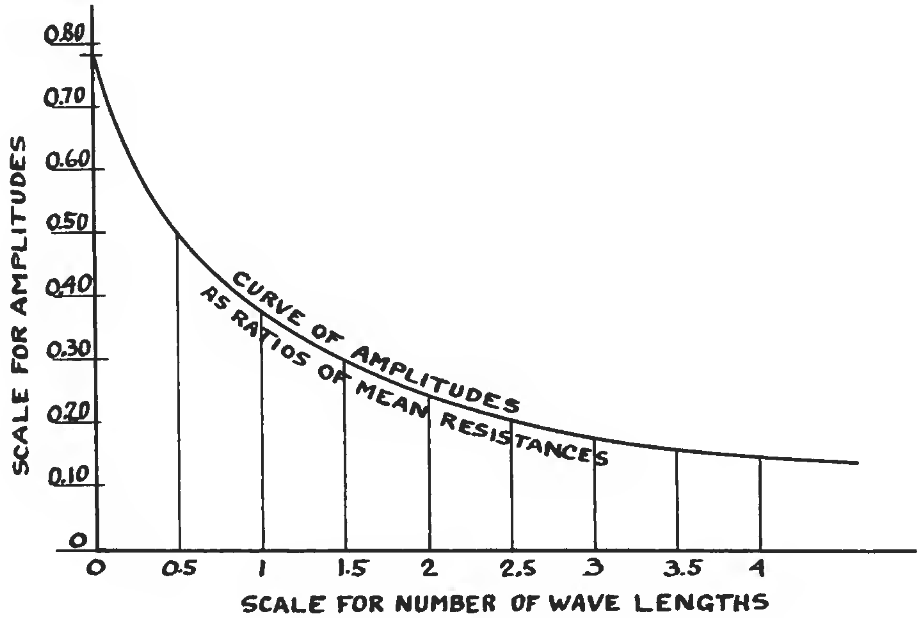


FIG. 31-A. CURVE OF AMPLITUDES, BASED ON NUMBER OF WAVE LENGTHS.

SHIPPING BOARD MODELS. Although the writer's theory agreed fairly well with the Froude experiments mentioned above, recent model tests have proved that the waves were not arranged symmetrically over the ship at a minimum. The distance from the stern to the last crest is always greater than that from the bow to the first crest, as mentioned under formula (50). While this fact does not alter the amplitude, the longer distance aft to the last wave crest reduces the speed-length ratio at which the minimum occurs. The differences are not large; however, the new formula for minimum developed from the Shipping Board tests, looks thus:

Minimum at

$$\frac{V}{\sqrt{L}} = \sqrt{\frac{1.0}{0.313 \times \sin a \times \cos a + (0.35 + n)}} \times 1.341 \qquad (54)$$

For maximum, the only change is that the term in parentheses in the divisor becomes $(0.85 + n)$. The constant $1.341 = \sqrt{1.0/0.555}$, *see* formula (50).

For the neutral speed-length ratios the term in parentheses becomes $(0.63 + n)$ and $(1.13 + n)$, there being always one $V/\sqrt{L}$ on each side of a minimum. Note that n is an integer here, 0, 1, 2, 3, 4, the half wave length incorporated in the parentheses for the maxima.

TABLE VI

Bow Wave Interference

Investigator	*Number of Wave Lengths on Ship*				
	0	1	2	3	4
1) Speed-Length for Minima.					
Froude 1877, 13.79 knots	...	1.113	0.857	0.722	0.636
Taylor,[23] Fig. 147	...	1.113	0.858	...	...
Havelock,[38] 350-ft. disturbance	1.51	1.200	...	...	...
Liljegren, 1899 and 1941	2.04	1.114	0.855	0.720	0.633
2) Speed-Length for Maxima.	0.5	1.5	2.5	3.5	4.5
Havelock	1.92	1.37	1.10	...	...
Liljegren	None	1.41	0.965	0.781	0.670
3) Amplitude for Minima.					
Froude	...	0.333	0.310	0.160	...
Taylor	...	0.525	0.315	0.165	...
Havelock	0.55	0.50	...	...	...
Liljegren	0.785	0.382	0.248	0.181	0.141
4) Amplitude for Maxima.	0.5	1.5	2.5	3.5	4.5
Havelock	0.755	0.712	0.333	...	...
Liljegren	0.500	0.305	0.210	0.160	...

Notes. All figures corrected for frictional resistance. The similarity of the Froude, Taylor, and Liljegren figures is quite amazing contrasted to the Havelock pure theoretical ramblings. The Taylor amplitude for minima at $V/\sqrt{L} = 1.0$, is evidently erroneous, as it leads to 1.3 at $n = 0$. Compare with Havelock's 0.55. The bow angles are varying for Liljegren, for others constant.

Example. To find the speed-length ratio and the amplitude for a given ship, $n = 2.0$, look under 2.0, and get 0.855 and 0.248. Always

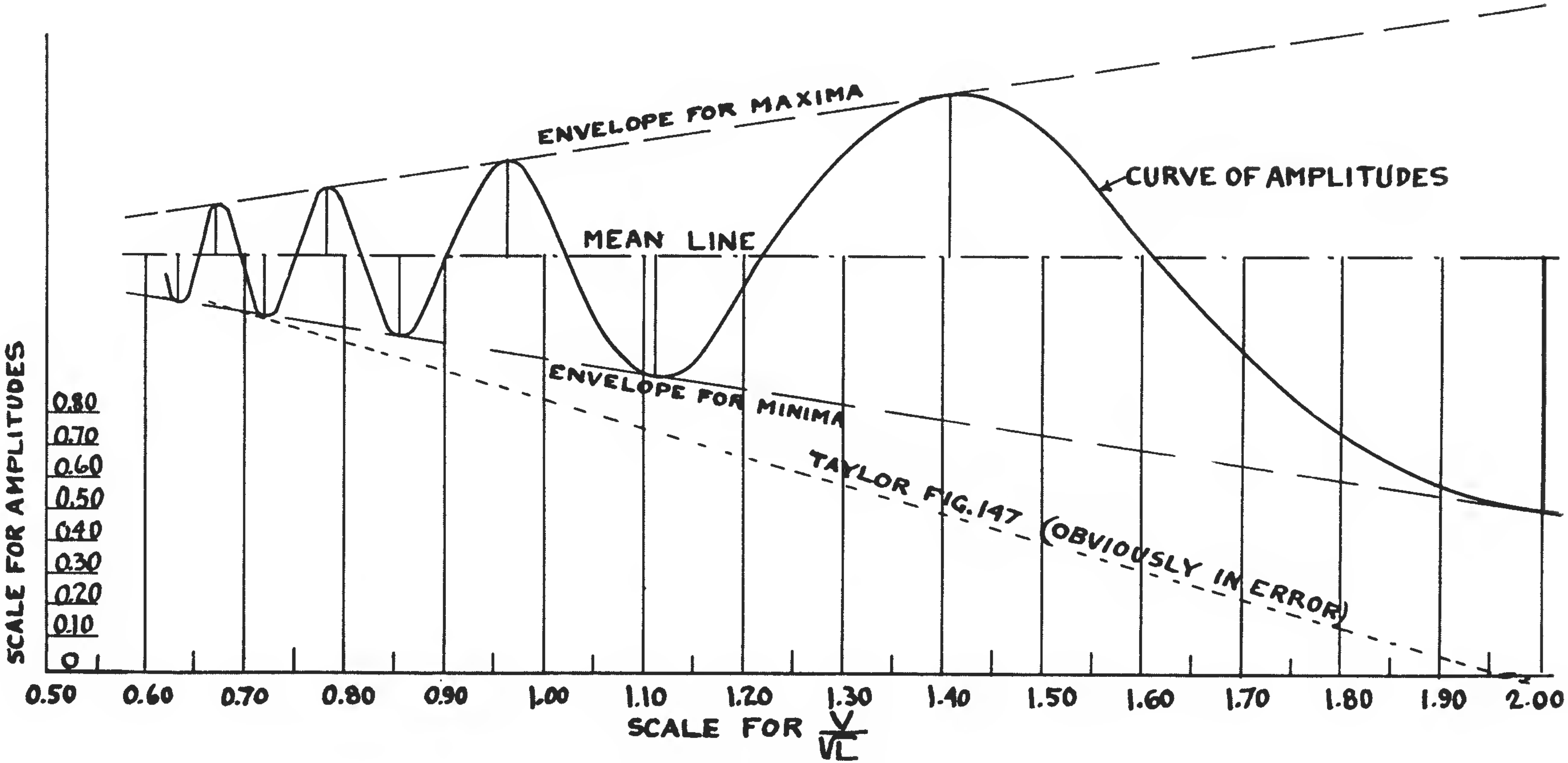


FIG. 31-B. CURVE OF AMPLITUDES, BASED ON SPEED-LENGTH RATIOS.

use figures for minima, leave maxima alone. The former are fundamental for *optimum vessels*, the latter for *"pessimum" vessels.*

The Curve of Amplitudes based on number of wave lengths on ship is pictured in Fig. 31-A, and based on speed-length ratios shown in Fig. 31-B. The former curve is hyperbolic, the latter straight lines, one below for minima, one above for maxima, the mean line in the middle. The curves are called envelopes, and the actual undulating curve of amplitudes is pictured in Fig. 31-B, from which the amplitude for any speed-length ratio can be measured off.

PROPULSION

The factors of propulsion have been described before (page 39), where it was explained how shaft horsepower is always higher than effective horsepower because the propeller cannot reach over 74% efficiency. There are, however, several other phenomena affecting the engine power, of which wake factor and thrust deduction factor are considered here, because there is always interaction between wake, thrust, and the ship. Cavitation, another phenomenon, is described under the Propeller Section, page 152.

Every vessel is accompanied by a forward flowing stream called *wake*, composed of at least three different motions of the water particles. The adhesion of the water to the surface of the hull, misnamed friction, causes a drag, the speed of which is highest at the midship section, where it amounts to just one-half of the ship's speed close to the surface of the hull. But the thickness of this *boundary layer* is very insignificant, only a few inches, and is the same for the same surface material—whether the vessel is big or little. Its speed is quickly dissipated by the water flowing in from the sides to fill the void aft of the ship. Over the disc area of the propeller, its speed is probably not more than 0.5% of the ship's speed, and decreases as the length of the ship increases. A model must always have a higher wake factor, as the ratio between the wake speed and the ship speed is termed.

The bulging form of the hull makes the water close to it *lag behind* the position of the undisturbed water farther out. This lag is zero at the keel, if straight, and maximum near the *LWL;* it causes a forward motion of the wake in addition to that of the boundary layer, and its part in the wake factor is the same for model and ship. The position of the

last bow wave at the stern largely affects the wake; if a crest over the propeller, the wake is increased in speed, if a trough, it is reduced; for the particles in a wave rotate in a forward direction in the crest, and in a backward direction in the trough. The orbital speed can never exceed $V \sin a$, the divergent speed at the bow, but for every wave length added on the ship's length the orbital speed is reduced as $n + 1$, n being the number of wave lengths. In other words, the orbital velocity is proportional to $V/\sqrt{L}$, and the resulting wake to V, but is additive when a crest is located over the screw, and subtractive when a trough is thus located. In the first case the wake factor w is increased, and in the second reduced in value. The orbital velocity decreases nearly as the cube of the depth below the surface, and the mean wake over a small propeller is greater than that over a large propeller, at the same speed.

To sum up, we see how complicated is the wake phenomena, too complicated for any theory that could result in a formula for w. Many such formulas have been tried by the writer, and found very wanting. The most conclusive experiment was made by Professor E. M. Bragg[39] in 1924; but his formula (proposed) does not take cognizance of the effect of the bow wave, nor of the higher wake of the model over that of the similar ship.

There is a difference between the American and the British method of measuring wake. The speed of advance of the propeller V_a through water is lower than the speed of the ship, the American formula being $V_a = V(1 - w)$, or $w = 1 - \frac{V_e}{V}$. The British formula is $V = V_e (1 + w_p)$, or $w_p = \frac{V}{V_e} - 1)$. As mentioned, w is termed *wake factor*, but Froude called w_p *wake percentage*. The relation is $w = \frac{w_p}{1 + w_p}$. From the mass of wake data, the only safe conclusion seems to be that the model has a higher wake than the ship, and that the fullness of the hull form, as measured by the prismatic coefficient C_p, increases wake as C_p^2, other things being equal. Twin and quadruple screws are less affected by wake than single screws. Besides Bragg cited above, Froude, Taylor,[41] and W. J. Luke[40] have made extensive wake experiments.

In any fluid whether water or air, a vessel can be moved forward

only by means of reaction from a fluid column moved backwards by a propeller. There are many kinds of propellers, such as side wheels, stern wheels, jets, and screw propellers, the last all but universal in deep-water craft. Of these the stern wheels show the lowest efficiency as measured by Output/Input = EHP/SHP, the screw propellers the highest. From Bernoulli's equations it follows that when the speed of the column of water is increased, the pressure inside is reduced and acts as a suction on the after surface of the hull below water. The suction has no balancing counterpart on the forebody, and evidently adds to the resistance of the vessel, and to the necessary push or *thrust* of the propeller. This addition to the thrust was termed *thrust deduction* by Froude—a perfectly absurd appellative, since it is exactly the opposite—an addition, not a deduction.

If the reduction of the pressure on the afterbody is expressed as a fraction t of the thrust T, then $T(1 - t) = R$, the total resistance of the towed vessel. The output or EHP is proportional to RV, and the input or THP to $TV(1 - w) = RV \times \dfrac{1 - w}{1 - t}$. The ratio $\dfrac{EHP}{THP} = \dfrac{RV(1 - t)}{RV(1 - w)} = \dfrac{1 - t}{1 - w}$. The latter ratio has been termed *hull efficiency* by Froude and often found to be over unity up to 1.08 or more. The output would thus exceed the input like a *perpetuum mobile* or even worse than that. Even for the best electrical dynamo the output seldom exceeds 96% but, granted 98% hull efficiency, t must always exceed w in value.

For instance, assume $w = 0.15$, then $\dfrac{1 - t}{0.85} = 0.98$, $t = 0.17$. Whenever hull efficiency exceeds unity, either w or t or both have been erroneously computed. According to Bragg,[39] using known propeller data by Taylor, the investigation of R. E. Froude leads to a propeller efficiency of way over unity, when even Froude could not get over 77%. The lowest $w = 0.01$ as given in reference 2, the highest $w = 0.53$ is given by Bragg.[39] The corresponding values of t are 0.01 and 0.54. The latter figure means that the resistance of the ship would be more than doubled by the action of the propeller. Taylor[23] states that Froude laid down the dictum that for twin screw vessels, $w_p = t$. From the relation $w = \dfrac{w_p}{1 + w_p}$ follows that $\dfrac{w}{1 - w} = t$. For $w = 0.15$, $\dfrac{1}{0.85} =$

$1 + t, t = 0.176$ very close to 0.170 estimated above. But for $w = 0.54$, $t = 1.18$, and the hull efficiency would be negative, an impossibility, and another dictum blasted.

Perhaps the most important refutation of hull efficiency greater than unity is the fact that warships, designed and tested with all the finesse available, never show a hull efficiency = 1.0. *U.S.S. Lexington*, for instance, with electric drive of the screws, possessed a hull efficiency of only 0.967 at 30 knots, as stated by Taylor.[2/3] w was then = 0.087, $t = 0.126$.

PROPELLERS

All propeller theories have been based on two erroneous assumptions:

1) *That the skin frictional resistance varies as* V^2, *the speed of the water over the propeller surface.* Our formulas show that the speed varies nearly as V, or $V^{5/4}$ to be exact, on account of the narrow blades.

2) *That the screw race is a solid column of water driven backwards by the propeller.* What the actual effective percentage of the disc area is, has never been discovered.

No wonder that all theories were at variance with reality.

The estimating and designing of a propeller to suit a given ship is a highly special line of Naval Architecture, developed by both Froudes, by Taylor, Schaffran, Dyson, and many others. Many of them experimented with model propellers up to 18 inches diameter as used by Taylor. From the experiments many useful propeller diagrams were made and those from two countries compared by Schaffran,[42] and found to agree very well. The diagrams are much more satisfactory than theories and formulas.

The screw propeller was first applied to ships by Joseph Ressel, an Austrian, in 1829, who perhaps got his idea from the screw of Archimedes. He was not successful, but in 1836 John Ericson, the Swedish inventor of the *Monitor*, and an Englishman, Smith, got better results. The screws of both were very long in an axial direction, that of Smith having two complete threads. In running aground, one-half of the screw length was lost, after which the Smith ship went faster,[43] and that led to narrower and narrower blades.

DEFINITIONS. The most important dimensions of a propeller are the diameter, the pitch, and the blade width. The *diameter* is the diameter

of a transverse circle touching the tips of the propeller blades. The *pitch* is the distance that the screw would travel in one complete revolution if fitted into a nut. The *width* of the blade is the greatest distance from the leading edge to the following edge. The developed area of a blade is the surface of its face, the blade area is the developed area of all its blades, the disc area is the area of the transverse circle. The pitch ratio is pitch ÷ diameter, the area ratio is developed area ÷ disc area, the mean width ratio is mean width of blade ÷ diameter, the blade thickness ratio is blade thickness at the axis ÷ diameter.

The projected area of a blade is the projection of its outline or contour upon a transverse plane perpendicular to the axis. The blade face is its after or driving surface, the opposite surface is called the back. The part of a propeller from which the blades stick out is called *hub*, or *boss*, as in a wheel, and it is keyed to the propeller shaft with one or more strong keys. The hub is held to the shaft by a big nut. Formerly, the hub was shaped as a can, and the nut was a nut without the least regard for its resistance. The writer was the first to form the nut as a conical tail to the streamlined hub, now universally used. Of late the hub has been sawed off and the tail continued on the sternpost.

There are right-handed and left-handed screws. A right-handed propeller turns clockwise when looking at it from aft, a left-handed turns counterclockwise when driving the ship ahead. The pitch angle is the angle of a triangle wrapped around a cylinder to represent the screw thread. It is the angle between the tangent plane and a transverse plane. In a true screw this angle is constant from root to tip of the blades, but in a variable pitch propeller, the pitch angle varies. When a propeller is working in water producing a thrust, it advances per each revolution a distance less than the pitch. The difference is called *slip* of the propeller, and is measured in percent of the pitch. It really means a shortened pitch and a smaller pitch angle: the difference between the two angles is called slip angle (*see* Fig. 32), which may be called a *lay-out* of propeller particulars. At the point O, is drawn a section of the propeller blade, face downwards, curved back upwards. The blade is supposed to rotate from left to right in the direction of the arrow, and in so doing pushes the water backwards. The reader is supposed to look at it from above. The triangle OPM represents the unrolled triangle, the angle POM equals the pitch angle θ (pronounced *theta*), MP = the pitch p, $OM = 2\pi R$ = the circumference of a circle with the radius R of the cylinder mentioned above. The section at O is taken at this radius R from the axis.

If the propeller was working without slip, it would push the water backwards the distance $PM = p$ for every revolution, but the water yields, and the distance becomes only SM, PS being the slip, and POS the slip angle a. Also $SM = PM(1 - \text{slip})$. The large triangle in Fig. 32 can also be said to represent velocities for, if multiplied by the number of revolutions per minute N, $OM \times N$ = circumferential velocity that varies with R, and $SM \times N$ = velocity of advance, con-

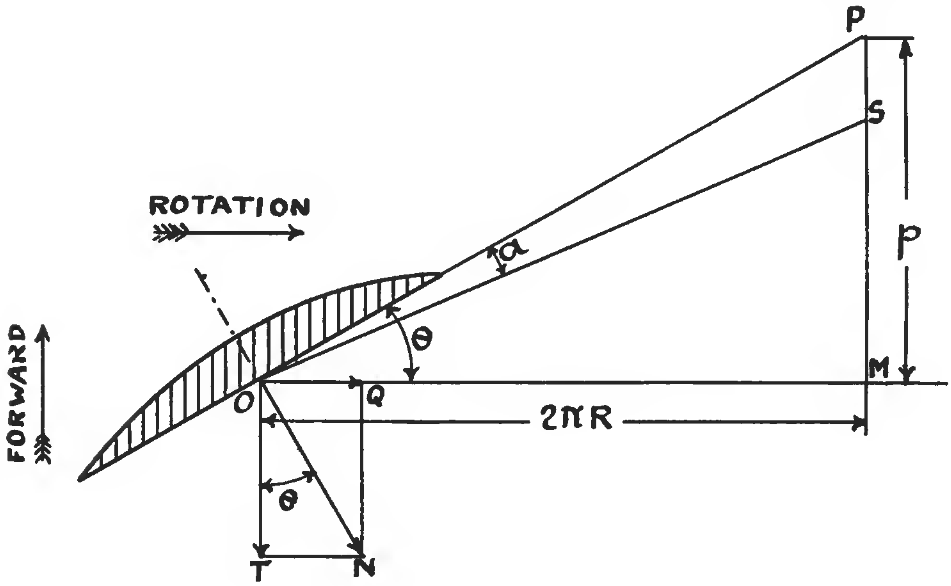


FIG. 32. PROPELLER LAYOUT.

stant if pitch be constant. In symbols the velocities are $2\pi R \times N$ and $p(1 - s)N$. The slip ratio is defined by $PS/PM = s/p$ which is a fraction but is usually multiplied by 100 and expressed as percentage.

From the triangles of speeds, Fig. 32, it is clear that the water particles strike the blade face in the direction SO, the resulting normal pressure is in the direction ON, nearly square to the surface. Because of the skin friction and the influence of the back of the blade, the normal pressure is not actually square to the face but is generally assumed so. The two components of ON are $OT = ON \cos \theta$, and $OQ = ON \sin \theta$, the former proportional to the *thrust* of one blade, the latter to the force on the blade resisting the turning moment or *torque* of the engine. It is practically impossible to estimate ON by a formula, hence the designer has to use diagrams, of which there are a great number published. Taylor[28] shows some excellent diagrams of four-bladed propellers of which photostats can be had at slight cost.

The diagrams are based on *true slip*, the slip of a model propeller advancing *before* the model hull in undisturbed water. Speed of advance is then $= V$. Behind a ship the propeller is greatly affected by the wake, the speed of advance $= V_a = V(1 - w)$. The true slip $s = \frac{np - V_a}{np}$, where n = revolutions per second, and V_a = velocity in feet per second. There is also an apparent slip, but this should never

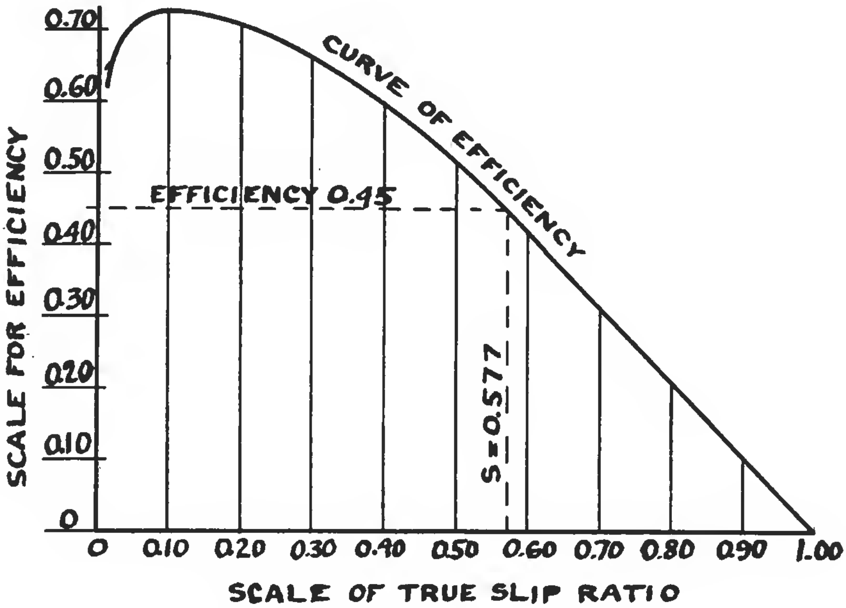


FIG. 33. TYPICAL PROPELLER EFFICIENCY CURVE.

be used in computations. The diagrams show, besides the thrust and the torque coefficients, the efficiencies of propellers with several pitch ratios p/d, and at all slip ratios. The efficiency e is measured by the ratio output ÷ input as usual, but this efficiency is for a propeller in free water, and the efficiency behind the ship is wanted. The output of the propeller behind the ship $= TV_a = RV \times \frac{1 - w}{1 - t}$, but the output of the free propeller $= RV$. In both cases the input $= SHP$, the shaft horsepower, and in the latter case, output ÷ input $= e$, the efficiency as measured from the diagram.

Hence in order to obtain the propeller efficiency behind the ship, e must be multiplied by $\frac{1 - t}{1 - w}$, and thus

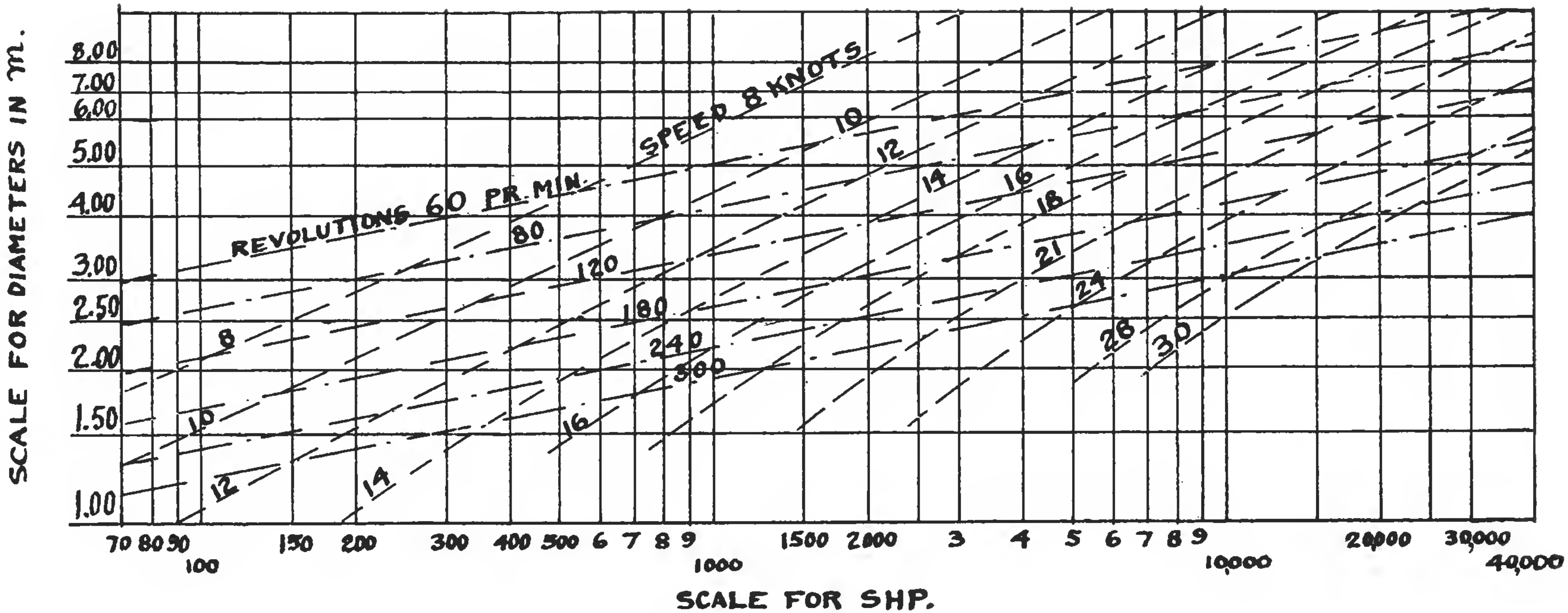


FIG. 34. OPTIMUM PROPELLER PARTICULARS.

$$\frac{EHP}{SHP} = e \times \frac{1 - t}{1 - w} \tag{55}$$

If the hull efficiency could be larger than unity, the propeller efficiency would be greater behind the ship than in free water against which possibility there are two main reasons:

1) The propeller must clearly work with higher efficiency in undisturbed water than in the turbulent wake behind a ship.

2) The true slip is always greater than the apparent slip in the ratio $\frac{1 - s}{1 - s_1} = 1 - w$, and as s_1 (apparent slip) is mostly close to the peak in the efficiency diagram, the actual propeller efficiency behind the ship must be lower (*see* Fig. 33). The peak is at 10% slip; suppose $s_1 = 10\%$, and $w = 0.53$, then $1 - s = 0.90 \times 0.47 = 0.423$, and $s = 0.577$. Both s and s_1 have been indicated in Fig. 33, their efficiencies being 73% and 45%. Yet the hull efficiency would be much greater with the higher slip, and with it the propeller efficiency, which seems hardly possible, since s is unaltered.

Possibly the error lies in assuming a gain from the wake, which reminds us too much of the farmer who was so strong that he could lift himself by his bootstraps. At the same time, the thrust horsepower *THP* is undeniably affected by the wake—it may be by only a fraction of the wake, or $THP = TV(1 - cw)$, c being a constant less than 1.0. Here is a suggestion for those model experimenters who disapprove the Froude dictum with its corollary of screw efficiency above unity.

Optimum Propellers. Since there are no reliable propeller formulas for propeller efficiencies, it is not possible to obtain the optimum propeller for a given ship by differential calculus, as in designing a ship. By making certain assumptions as regards wake, pitch ratio, slip, and efficiency, it is possible to make a diagram from which the optimum propeller can be picked (*see* Fig. 34). It is a logarithmic chart based on shaft horsepower, propeller diameter scale to the left in *meters* (1m. = 3.28 ft.). The sloping lines indicate revolutions per minute, and speed in knots. Wake is assumed 0.15, $p/d = 1.0$, $s = 25\%$, $e = 67\%$, and three elliptical blades mean width ratio $= 0.20d$.

In using the diagram, follow the *SHP* for the screw to where it intersects the speed line, read off the diameter to the left by following the horizontal line from the intersection, and the revolutions from the sloping line, usually by interpolation. The revolutions should be compared with formula (13), page 51, and possible adjustments made. The elliptical blades should have their maximum width at 70% of the

radius, because the frictional skin resistance varies as the radius at each point, not as the R^2, as usually assumed.

CAVITATION. Besides wake and thrust deduction, there are other losses affecting the engine power of a ship. The most difficult to overcome is cavitation, resulting from too high revolutions. A smaller loss comes from the *inclination* of the stream paths to the plane of the propeller.

A two-bladed propeller would let the water column slip easily through its plane, hence could not cause cavitation except at much higher revolutions than practically permissible. But as the number of blades is increased, there comes a speed when the centrifugal and other forces cause the propeller to spin in near vacuum as a top without pushing back any race at all, thus producing no thrust or a very unsteady thrust. Cavitation was first observed in 1894 by Mr. Sydney W. Barnaby in the British destroyer, *Daring*. He came to the conclusion that cavitation began when the thrust pressure on the face of the blade was slightly over 11 lbs. per sq. in. of the projected area. But at the high speeds of the blades through the water just before cavitation begins, the blades cut through like a knife, and projected area could have little effect, whereas the thickness of the blade must count heavily. Taylor[23] states the thrust pressure on *U.S.S. Lexington's* screws reached 22.4 lbs. without apparent cavitation.

The fact seems to be that the water can follow along the blade sections only up to a certain speed, and beyond that leaves the surfaces of both face and back, causing eddies and loss of thrust-producing surfaces.

Schoenherr[44] has developed an entirely different theory of cavitation wherein he states that cavitation starts when the total energy per unit volume in the stream lines meeting the propeller is completely converted into kinetic energy. This assumption seems to be contradicted by the fact that recent photographs show the hole in the water made by the propeller during cavitation. But he gives interesting examples which tend to show that airplane sections near the tips of the blades lead to earlier cavitation than ogival sections. This agrees well with the cleavage phenomenon stated just above.

Taylor[45] describes some experiments with cavitation in the pressure tunnel at the Washington Model Basin, using 8-in. model propellers, from which it is clear that cavitation starts when the hydrostatic pressure around the screws is reduced to one-fourth of the atmospheric

pressure in feet of water. Unfortunately, neither the revolutions nor the thrust were given in the paper for converting the result to actual cases.

It can be shown, however, that the paths described by water particles around a submerged body are circular arcs, because the forces tending to fill the cavity always produce centripetal acceleration like the string of a sling or pendulum. This fact indicates that the sections of a propeller blade should be given sinus-curved form exactly as trajectories of ships, in order to delay cavitation.

Twin Screws can never reach the efficiencies of single screws, because the stream paths around the screws are unsymmetrical and meet the propellers at greater mean inclinations than those around a single screw. The loss of efficiency and thrust is $1 - \cos^2 a$, where a is the mean angle of incidence.

TUNNEL SCREWS. Propellers for shallow draft vessels fitted in tunnels, instead of under the hull proper, are called tunnel screws. Such screws may have a diameter up to twice the draft of the vessel, and are thus only partly immersed, which causes some loss of efficiency. Partly immersed screws were first used by Mr. Hicks of New London, Conn., who called them *vane wheels* and placed them just aft of the boat's transom. Their efficiency was very low, not over 50%, since most of the race was thrown sky-high as spray. To curb this unpleasant feature, conical or cylindrical splash screens were fitted above the screws, which improved the efficiency and might have led to tunnels.

Tunnels are gouged or cut out in the bottom of the vessel, begin some distance ahead of the screw, rise in a fair curve over the propeller, and then sink down to the water line at the stern (*see* Fig. 35). Two or more rudders are fitted into each tunnel because the ship's steering qualities must be the best, both ahead and backing. The transverse sections of the tunnels are usually circular arcs, with rounded corners where the bottom of the vessel is joined. The maximum tunnel radius is about 3% larger than the screw disc circle.

When the propeller starts working, the water in the tunnel is drawn aft and violently splashed about. After a while, the whole tunnel is filled with water lifted up from below. Of course, the lifting process detracts a great deal from the power and efficiency of the propeller—already perhaps under 40% at the start. The pitch ratio of tunnel screws is often as low as 0.40, and the lower, the less the efficiency.

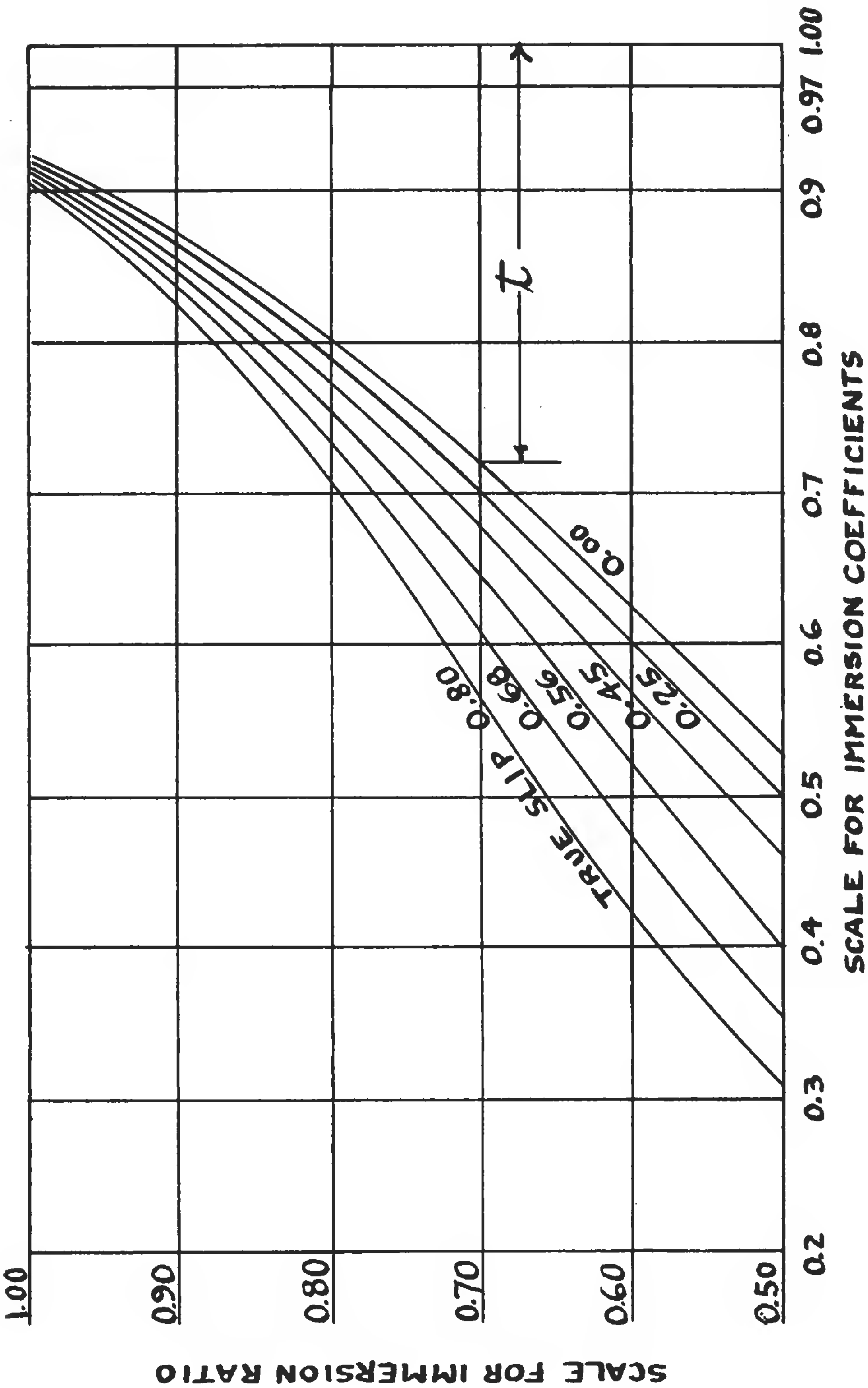


FIG. 37. PITCH RATIOS AND EFFICIENCIES.

At the same time the slip is very high, which, as seen from Fig. 33, means *a priori*, a low efficiency. Then, to crown it all, the propeller may be only half immersed when at a standstill, although usually it is more. Fig. 37 gives a picture of the *immersion coefficients of efficiency*, each curve for a certain slip ratio, the horizontal lines for the immersion ratio, the vertical lines for the coefficients. For instance, at 25% slip and 50% immersion the coefficient = 0.5, at 56% slip it is slightly less than 0.4.

In other words, the thrust or push of the propellers is reduced, as the immersion coefficients indicate, from its value when the screws are fully immersed and covered by at least $0.2d$ of water to prevent air from being sucked in. In the latter case, the thrust deduction factor is assumed to = 0.03, a minimum for long easy flow lines to the propeller. At every immersion, the distance from the right end of the diagram to any of the curves equals the thrust deduction factor t at the particular slip and immersion. The factor t is thus equal to 1.0 − immersion coefficient, and increases slowly with the slip ratio.

As stated before, it is usually possible to design an *optimum* propeller for a deep-water craft but not so for the low speeds of river towboats. For similar craft, and indeed for all tugs, optimum screws would be far too big, and their revolutions far too low for their engines. All the designer can do is to fix the largest screw and the lowest revolutions for each case, and from the immersion coefficient obtain the screw of highest efficiency and shaft horsepower constant. The towing speed should be regulated by the average water depth, and the pitch of the screw is dependent on the speed of advance and the revolutions.

For every pitch ratio there is a slip that offers the highest efficiency possible, as shown by Fig. 36 (folding plate), based on the complement of the slip ratio multiplied by the pitch ratio a, or on $a(1 - s)$. Curves for five pitch ratios are drawn, the lowest 0.50, the highest 1.05. Suppose our data give a slip ratio $s = 0.40$, and a pitch ratio $a = 0.50$, $a(1 - s) = 0.30$, we read off directly from the curve for $a = 0.50$ that efficiency e is equal to 0.47. This value of e must be multiplied by the proper immersion coefficient from Fig. 37 to get the tunnel screw efficiency. By trial and error with several screw diameters and revolutions, the best possible screw is finally found.

Of course, Fig. 36 can be used for deep-water screws as well, whether the screws are fully immersed or part out of water when the ship runs in ballast.

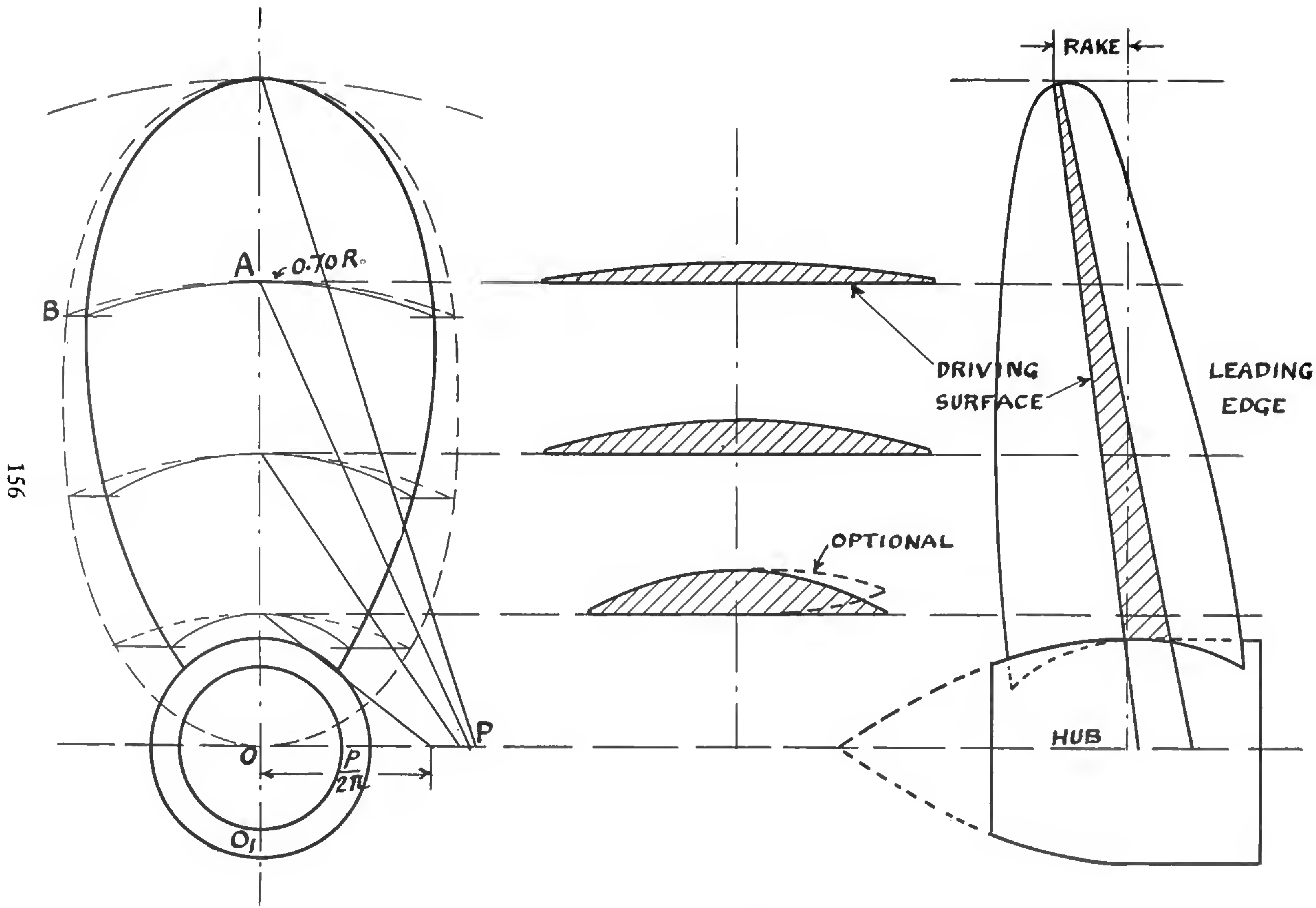


FIG. 38. PROPELLER PLAN (*see* SCHOENHERR, FIG. 16).

Multiple Screws. From the foregoing it is evident that if the immersion can be reduced and the pitch ratio increased, the efficiency of a tunnel screw can be greatly increased, even for small changes in immersion and pitch ratio. In deep-water ships, quadruple screws have been very efficient, and triple tunnel towboats have shown increased towing efficiency over two-tunnel boats; but to break up the bottom by four tunnels would be not only too costly but also would increase the wetted surface considerably just where the screw race causes extra skin resistance. To gain the advantage of many small screws in only two tunnels, the writer has suggested fitting two or more propellers *to each shaft*. In this manner higher pitch ratios can be used, resulting in higher propulsive efficiency, and, in addition, lower revolutions might contribute to the same end. The principle of several screws on the same shaft has been applied to torpedoes for over fifty years with complete success, as proved by their speed of more than 40 knots.

Propeller Plans. In making the propeller plan, one is confronted with the difficulty that the helicoidal face of the blade cannot be expanded onto a plane surface, but the error resulting therefrom can be made so slight as to be negligible. The main difficulty is shaping the sections and giving them their correct twist or pitch angles. The contour of the blade and the sections are laid down (projected) on a transverse plane cutting the axis at right angles. A longitudinal plane is only used to show the side view of the hub, the tail nut, and the blade.

To obtain the pitch angle at each section set off from the axis O, Fig. 38, on the horizontal coordinate, the distance OP = pitch $\div 2\pi$. Then a line joining P and any blade section on the vertical coordinate, as at A, makes an angle OAP equal to the pitch angle of the blade face. If the pitch be constant, all the lines from the sections meet at P, otherwise there will be a P for each section. Usually the pitch is increasing from the axis to the tip of the blade, and the mean pitch is measured at $0.7d$, where the center of pressure is supposed to be located (true only for a rectangular blade). With the expanded contour drawn on the plan, and thickness at the center line of the blade obtained from the Registry Societies, the sections can be finished. The distance from the edge of the sections to the center line gives the side view of the blade or its projection on the longitudinal plane through the axis.

But the projected outline in the other plan is not so easily drawn on the plan. Swing first the section, say at A, to a horizontal position, then draw the circular arc through A with the radius OA. On this arc set off the width of the blade projected on the center line (the vertical coordinate) when the section is in its original position. The width of the plane is not measured along a horizontal line through A but along a circular arc with the radius O_1A which is equal to $AP/\cos OAP$, and is a very close approximation to the elliptical arc formed by a cylinder cutting the blade face at A. If a horizontal line is drawn from the intersection of the larger arc and the developed contour of the blade as at B, the line should meet the projected outline on the smaller circle.

The diameter of the hub is about one-fifth of the propeller diameter, its length a little more than the side projection of the blade at the hub determines. When there is nothing aft of the rudder as in twin screws, the hub and the nut should taper gradually from the forward end to the point of the nut, with the shaft bearing streamlined into the hub. When there is a rudder aft of the propeller, the hub can be made more cylindrical and a tapering tail fitted on the rudder post. For very large propellers the nut is separate and of the usual form, the tail of conical shape being hollow.

Usually, the propeller blades of a single-screw ship are given a rake aft, sometimes as much as 2 inches per foot of the radius. The rake reduces the thrust deduction factor and increases efficiency of propulsion. In the propeller drawing, the rake is shown in the side view.

PROPELLER BOOSTERS. It has long been known to propeller experts that much power is lost in the turbulent flow to the screw, and in the rotation of the screw race. In 1905, the German, Dr. Eng. Wagner, hit upon the idea of the *contra-propeller*, a misnomer since it is no propeller at all but consists of several blades fixed in position aft of the screw, and shaped somewhat like turbine guide blades. From the contra-propeller the water particles are issued in a straight axial direction. Still earlier, Thornycroft[46] had built a turbine propeller enclosed in a cylindrical casing, with guide blades and a long fixed tail aft of the screw. Of late, much has been written about the *Kort nozzle* that purports to force the water particles to meet the propeller squarely to its disc in the same manner as the water meets a model propeller placed before the ship model.

Now, if the water is forced to alter its direction, there is always

created a resistance, as, for instance, by the bow wave. With guide blades aft of the propeller, there is additional resistance as a drawback (literally). Undeniably, a gain in efficiency must be credited to these and similar inventions but, as a new screw is fitted every time, it is hard to determine how much gain from the screw and how much from the invention. Besides the extra resistance, such a combination always works very badly when backing, and is liable to be damaged by ice, weeds, and driftwood.

Herbert Hoover, LARGEST DIESEL TOWBOAT IN THE WORLD. SHP, 2,200. OWNERS, FEDERAL BARGE LINE; DESIGNERS, GIBBS & COX; BUILDERS, DRAVO CORPORATION. *Courtesy of the Owners.*

TOWBOATS

On most of the great rivers of the world, shallow-draft vessels called tugs or towboats are towing other vessels called barges. As a rule the tug is ahead of the barge, but on the western rivers of America, the barges are pushed by the tugs—which should really be called push-boats but are termed towboats by all concerned. Formerly, these tow-boats were built as side- or stern-wheelers, their boilers fired by wood or coal. For the last 15 years, most towboats have been fitted with Diesel engines and tunnel screws on account of their much higher towing efficiency, which is only 0.15 in the best stern-wheelers with Diesels but reaches 0.30 for tunnel twin-screw Diesels, and up to 0.60 for multiple screws.

As in deep-water craft, there is an optimum combination of shaft horsepower, speed, and revolutions that result in the highest possible

towing efficiency per *SHP*. But unlike deep-water craft, towboats are designed for the smallest possible displacement, just enough to carry hull, outfit, and machinery on a given draft. The limited draft necessitates great beam, and as stability and seaworthiness do not figure in the design at all, the beam is often made one-third of the length in small towboats, and not less than one-fifth in the largest boats of 200 ft. in length.

The maximum towing efficiency is usually reached at a speed-length ratio of 0.45, at which the wake factor $w = 0.21$, the thrust deduction factor $t = 0.31$, the hull efficiency = 0.875, the true slip $s = 56\%$, and the propeller efficiency = 0.32 for an immersion of 0.84. These values are valid for towing in water the depth of which is 50% more than the draft of the towboat. Formula (41) tells that the perilous depth of water at a speed of 6 knots (6.9 miles) is 10 ft., hence the draft of the towboat would be 6.67 ft. for the values of the maximum towing efficiency given above.

When stability does not figure in the design, it would be possible to assume the Law of Similitude valid for river boats in general. The weights of the superstructures, and their windage, the greater weights of the hull per foot of length, and of the machinery per *SHP*, make it necessary to allow more beam per length in a small vessel than in a large one. In apportioning *SHP*, the higher skin resistance of the smaller vessel, per square foot of wetted surface, must be taken into account. On the other hand, the draft of the smaller vessel can usually be larger in proportion to length, and the wetted surface per ton of displacement smaller—distinct advantages, since skin resistance plays a large rôle at the low speeds of towing. From careful analysis of several towboats whose particulars and lines have been delivered by their owners, Fig. 39 has been compiled and delineated by the writer. It is based on the *net push* that can be expected from the engine at a towing efficiency of 0.30 for twin-screw tunnel towboats. The net push equals the thrust by the screws less the thrust deduction and less the resistance of the towboat itself. The resistance is greatly reduced by the wake of the towed barges, estimated as 25% of the speed of the tow, or $w = 0.25$. The speed of the towboat through the surrounding water is thus only $0.75V$, its skin resistance $R_f = 0.56V^2$, and its pressure resistance $0.32V^4$, V being the speed of the tow through undisturbed water, in knots.

This reduced speed must be used in connection with the towboat's

own wake = 0.21 (*see* above), and the true slip s. It all adds up to quite an appreciable gain for a towboat pushing the tow as against a tug towing its tow, but, naturally, pushing cannot be used except in fairly smooth water, and even then, the towing heads that take the bumps are strongly maltreated.

On the rivers, speed is always counted in miles. Towing efficiency equals push in pounds times speed × 0.00266 divided by shaft horsepower, or

$$\text{towing efficiency} = \frac{PV \times 0.00266}{SHP} \tag{56}$$

As mentioned, towing efficiency = 0.30 in Fig. 39, but can be bettered to 0.60 by multiple screws on the same shaft.

PADDLE WHEELS

Owing to their heavy weight, low revolutions and low efficiency, paddle wheels are now almost completely superseded by screw propellers in tunnels, but a description is here included for the sake of completeness. Paddle wheels exist as side wheels and stern wheels, the latter mostly built as an ordinary wagon wheel with paddles on the spokes. Side wheels are now always fitted with feathering paddles, or paddles that are made to revolve about an axis in their own plane as the wheel itself turns around. In this manner, the paddles are placed almost vertically in the water, thus increasing their grip and their efficiency.

The feathering mechanism consists of a system of links and rocker arms moved by an eccentrically placed tap, around which a sprocket turns. There is one link for each paddle, all freely movable except one that is fixed in the sprocket to make the links keep their relative positions. The mechanism is very intricate and heavy, so much so that a feathering wheel is almost twice as heavy as a radial wheel of the same diameter.

The idea behind the feathering is to avoid shocks and loss of efficiency every time a paddle enters or leaves the water. To effect this, the paddles in the water (mostly three) are directed towards the top of the wheel circle, but this construction is correct only if the wheel worked without slip—an impossibility. In that case, the resultant of the wheel speed and the ship's speed would be directed toward the

top of the circle. Such a construction takes no heed of the ship's own bow waves, nor of the slip of the paddles that lowers the water surface near the wheel, as Mr. J. L. Thornycroft[60] was the first to mention. The relative water velocity is thus no longer horizontal but dips slightly toward the wheel as if going over a low weir. As soon as the water is gripped by the paddles, it moves along with the speed of the paddles, which means that the leaving paddle must be directed toward the top of the wheel circle in order to avoid shocks. But to avoid shocks when entering water, the direction of the paddle must be set *forward* of the top of the wheel circle, for here the influence of the slip is felt.

The resulting construction is shown in Fig. 39-A. A is the center or axis of the wheel, B the lowest point on the wheel circle passing through the pressure center of the paddles, C is the highest point, toward which the leaving paddle at D is directed as shown. The length of the rocker arm DF is usually $0.6h$, the height of paddles. Usually $h = 1/5$ of the outside diameter of the wheel. The point F is the first spot on the pitch circle along which the ends of the rocker arms must travel as the wheel revolves. To obtain a second spot, at the entering paddle, the initial velocities and their resultant must be laid down in amount and direction.

While the leaving paddle at D is assumed to be at the level of the water surface undisturbed, we know that G is below that level, and that the velocity direction is not horizontal. Let GH represent this velocity, and GK the tangential velocity $2\pi RN$, then GP is the resultant velocity in feet per second, and also the direction of the entering paddle, set to avoid shock. Draw GR at right angles to GP, measure off the rocker arm length DF, then R is the second spot on the pitch circle whose radius is equal to that of the wheel circle R. Lay off the common center of F and R at E, then AE is the eccentricity of the pitch circle.

Sometimes the lowest paddle at B cannot be made to stand upright or in a vertical direction, but the slight difference does not matter for the efficiency of the wheel.

In order to obtain the magnitude of the velocities at G, a slip of 20% may be assumed. In other words, if V = velocity of ship in feet per second, then $2\pi RN = 1.25\ V$. But the water is, as mentioned, sucked into the wheel race with a velocity equal to one-half of the slip, according to the findings of most propeller experts, hence $V_a =$

1.125 V, and the true slip is only 10%—quite the opposite to the propeller slips. V_a is no longer to be set off horizontally from G but at an angle of 7.5° when the apparent slip is 20%, slightly increasing with the slip. The point G should be taken 0.04 D below the undisturbed water surface WL.

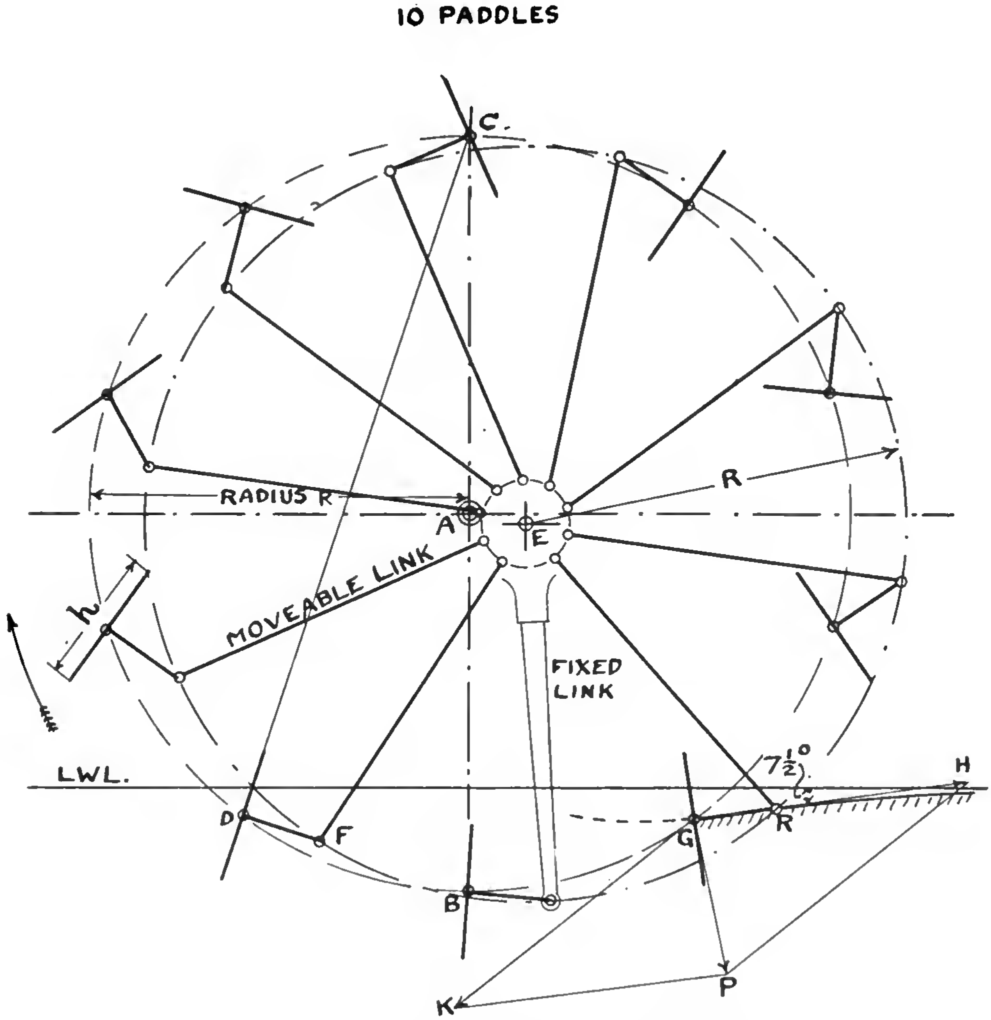


FIG. 39-A. PADDLE WHEEL CONSTRUCTION.

From the trochoidal wave theory the water particles move forward in the wave crest, and backward in the wave trough. Clearly, the wheel should be placed over the wave crest to get the most out of the wave motion. Note that while ocean waves are not formed as trochoids, the bow waves of ships are so formed as the writer has proved in an article written in 1898.

The position of paddle wheels is usually near the middle of the *LWL* of the ship or slightly aft of it. A wave crest there means a speed-length ratio of 0.75 but if the Optimum $V/\sqrt{L} = 0.72$ is to be used, the wheels should be placed at 0.40 *LWL* from the stem, instead of at 0.50 *LWL* with $V/\sqrt{L} = 0.75$. A compromise may sometimes be best.

Dimensions of Paddle Wheels. The dimensions of the wheel and its paddles allow a great variety of proportions, as long as the thrust of two paddles is greater than the resistance of the ship—how much greater depends on the efficiency of the wheel which is seldom more than 60%. The wheel diameter D is dependent on both the *SHP* and the freeboard of the vessel, but a first approximation is

$$D = 1.50\sqrt[3]{0.5\ SHP} \tag{57}$$

The revolutions per minute are 30 for big wheels, up to 40 for small ones, the depth to the lower edge of the paddle $= 0.23\ D$ below the undisturbed *WL*, the paddle depth $h = 0.18\ D$, the paddle length $b =$ Beam/4, and all to suit $R/2 = 5.7\ hb\ (2\pi RN - V)^2$, where $N =$ revolutions per second, $V =$ ship's speed in feet per second. Three paddles are assumed to be in the water at one time.

TRIAL TRIPS AND THEIR ANALYSIS

The purpose of a trial trip is to find out the connections between speed, engine power, revolutions, water and fuel consumptions of main engines and of the auxiliaries, the workings of anchor and steering machinery, the cruising radius, and similar data for the owners and the builders.

The ship should be in first class condition, bottom newly painted, the machinery so far as possible "run in," compasses adjusted, etc. The most useful trial trips include thrust measurements, but such are seldom made. So far as known, the first thrust measurements were made by the late W. R. Eckhart, who at the time was superintendent of U. S. Navy Yard, Mare Island, Cal. He wrote an excellent paper[16], unfortunately received too late for discussion at the meeting in London. The trial vessel was only a 50-ft. steam launch, it had to be small to measure the thrust by appliances then at his disposal.

Other famous thrust measurements have been made on the German cruiser *Mainz*[47], and U. S. destroyer *Hamilton*[25]. These trial trips have

given us such an insight into the conditions affecting ships at sea as could not otherwise have been obtained, as well as in the differences of model and ship.

While models are tested in still water without wind, there is always some air resistance if the model is hollow and not covered by some kind of smooth decking. Ships, on the other hand, have to contend with wind, waves, and water current, leeway and bad steering. To make accurate conversions, either the model data must be stepped up to the level of the ship, or the ship data stepped down. In any case, agreement is hard to achieve.

The course over which the trials are made must have a certain length, a depth of water at least 8 times the draft of the ship, *see* formula (41), and must be reasonably free from water currents. The latter is, of course, out of the question in tidal waters, hence several runs are made in opposite directions, usually four, and first, second and third means of the speed taken. This speed is very near the actual mean speed of the ship. In the same manner, the mean horsepower and revolutions are found.

The means are accurate only if the runs are made at the same speed, and the tidal currents vary from hour to hour according to a fair curve predicted by U. S. Coast and Geodetic Survey. This subject is presented in excellent form by Captain H. E. Saunders[25], to which the reader is referred.

The speed through water is checked by pressure and resistance speed logs, and compared with the mean speed from the trial result. It is easy to see what an elusive thing an accurate speed estimate really is, but perhaps no more so than the corresponding speed of the model. If thrust measurements have been made, and propeller characteristics are known, slip s, wake factor w, thrust deduction factor t can all be estimated, and search made for possible cavitation. If correctly estimated, t should always be greater than w, and w greater for the model than for the ship, unless the latter's underwater surface is rougher than that of the model. In converting from the model to the ship, the skin resistance of both should be estimated by the writer's formulas, and pressure resistance of the model corrected for wall effect if the model was tested in the Washington Experimental Basin or equivalent.

Air resistance calls for another correction, but by far not as large as ordinarily assumed. If a hollow model, its air resistance can be taken

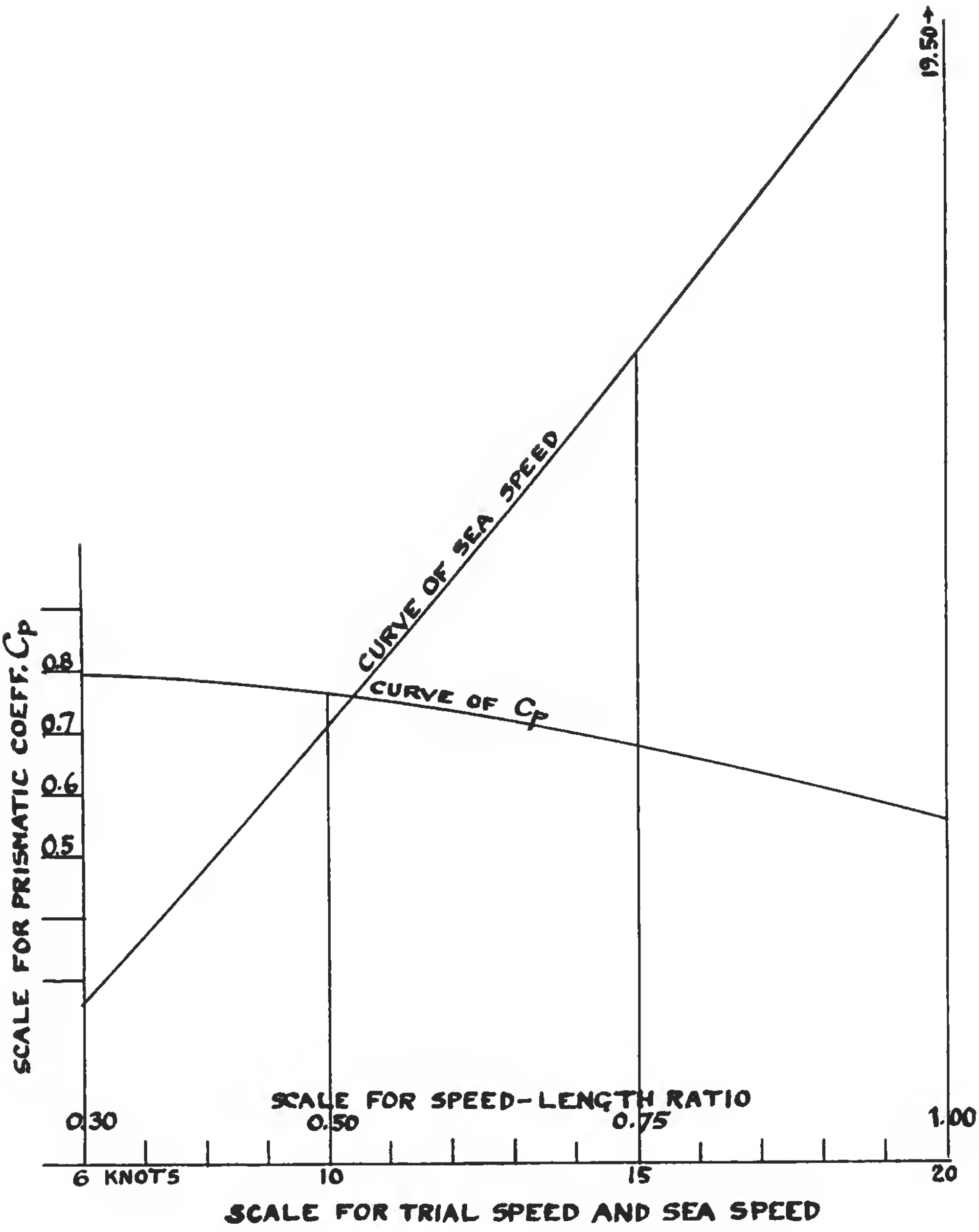


FIG. 40. SEA SPEEDS AND TRIAL TRIP SPEEDS.

as $0.0070AV^2$, *see* formula (40), where A = beam times freeboard; otherwise use formula (40). In the case of the ship, formula (40) is used for headwinds, but is increased 50% for a warship, and 30% for a cargo ship, at an angle of 30° between the apparent wind and the ship's course. The area A is the transverse area of the ship and the

bridge or superstructure above water. Taylor[23] states that $A = 0.5B^2$.

The method described above for one single speed can be applied to any number of speeds, but here a new factor must be reckoned with—the efficiency of the ship's machinery. However, if *SHP* and thrust is measured, this efficiency is eliminated. By trying each vessel at a number of speeds up to the highest, curves of *SHP* and V, *SHP* and T, or many similar combinations can be plotted as diagrams. One of the most important diagrams is the actual fuel consumption per shaft horsepower, for the main engines and for the auxiliaries. From this diagram, the cruising radius at full speed, and at half speed or less, is estimated.

Another useful coefficient is shaft horsepower divided by speed times displacement, or SHP/VD. It gives a rough estimate of the goodness of the entire combination: hull, machinery and propellers, especially within the same speed-length ratio.

Sea Speed versus Trial Trip Speed. Wind and weather, pitching and rolling, screw immersion when in ballast, fouled bottom, etc., contrive to make the average sea speed a good deal less than the trial trip speed with the same engine power. Fig. 40 gives an indication of the average loss for a 400-ft. ship at the speed-length ratios, and the prismatic coefficients shown. Investigations by U. S. Shipping Board[48] tend to show that 0.3% sea speed can be added for each 100-ft. increase in length, and 1% can be deducted for each 0.10 added to the prismatic coefficient, as for instance from $C_p = 0.60$ to $C_p = 0.70$.

STRENGTH OF SHIPS

As long as ships were built of wood, the strength question was never critical, for if too lightly scantlinged, it was easy to clamp on more wood. There was, however, a certain length of ship that could not be exceeded, and the sailing ship, *Great Republic*, Donald McKay's masterpiece, was already, with her length of 300 ft., far too heavy for profitable service. Even the first iron ships were mostly built to a rule of thumb until William John[33] in 1874 suggested some scantling rules that are in use to this day, even if altered and improved.

Mr. John calculated the strength of a great number of merchant vessels of different sizes, on the assumption that the longitudinal bend-

ing moment was equal to displacement times length divided by 35, or

$$\text{bending moment} = \frac{\text{W.L.}}{35}$$

with vessel poised amidships on the top of a wave of her own length. The maximum tension on the upper works he found to be 1.67 tons (3,740 lbs.) per sq. in. for a 100-ton (burthen) ship up to 8.09 tons (18,100 lbs.) per sq. in. for a 3000-ton vessel. The length of the latter ship was about 350 ft.

Dr. Bruhn of Det Norske Veritas also investigated the same stresses, and found for a 100-ft. ship 4.20 tons (9,400 lbs.) per sq. in., for a 350-ft. ship, 7.90 tons (17,700 lbs.) per sq. in., and for a 700-ft. ship, 9.90 tons (22,200 lbs.) per sq. in.

Now, all these vessels were built of the same material, and should be able to stand the same stress—Why then is there such a great difference between the smallest and the largest ship? One reason is the allowance for corrosion, which is the same for all sizes hence, naturally decreases the stress of the little fellow. Another reason lies in the fact that small vessels meet waves of their own length much more frequently than do large ships. A third reason is supplied by the fact that small vessels as a rule rest on the bottom at low water when in harbor. But all three together cannot explain the great differences in the stresses.

Thus the main reason must be that the bending moments are erroneously computed when the height of the wave is assumed to be $1/20 \times LWL$. Take, for instance, the 1,000-ft. liner *Queen Mary*. The height of her wave would be 50 ft., but such waves are never found in the open sea. The vessel's actual draft is 38.83 ft., the stress on her upper works amounts to 9.32 tons (20,850 lbs.) per sq. in. In order to meet this strain safely, the upper works were made of high-tensile steel, according to a paper by S. J. Pigott[49]. Now if the height of the wave is assumed to be equal to the draft, the stress would be 7.25 tons instead of 9.32—quite a sizeable reduction.

The bending moment $W.L./C = L^3B/C$ if the height of the wave is taken as $1/20 \times L$, but $= L^2BD/C$ if wave height equals draft. The immense influence of L in the first equation is evident.

Actually, the bending moment consists of the bending moment in still water added to the bending moment among waves. The former is quite appreciable; some 600-ft. long Great Lakes ships hog as much as 8 ins. fully loaded. Several of them have broken in two during great

storms, necessitating stronger scantlings. *Hogging* in this connection means that the vessel's deck is bent upwards amidships like a hogback; *sagging* means that the deck is bent downwards like a horse's back.

The fundamental equation for longitudinal strength of a beam is

$$p = \frac{M}{I/y} \tag{58}$$

p = stress per square inch
M = maximum bending moment in foot-tons
I = moment of inertia of midship steel section in feet^2 $\times$ square inches
y = the distance of the upper deck stringer from the neutral axis, in feet

The bending moment M = W.L./C as mentioned before; I/y is termed *modulus* of the section. When multiplied by the maximum allowable stress per square inch, p_{max}, the modulus is termed moment of resistance of the section, and is equal to the bending moment,

$$M = p_{max} \times \frac{I}{y} = \frac{\text{W.L.}}{C} \tag{59}$$

The modulus is now fixed by the International Convention to be equal to $f \times D \times B$, where f is a factor obtained from the Convention's Table (*see* U. S. Load Line Regulations), D = molded draft, and B = molded beam of the vessel.

The moment of inertia is calculated from mathematical rules; for a hollow square, outside height = H and inside height = h, $I = \frac{H^4 - h^4}{12}$, and its modulus = $\frac{H^4 - h^4}{6H}$ is nearly equal to that of the midship section of a cargo ship.

To estimate the moment of inertia I of the midship section, the areas of all continuous steel members below the strength deck are first computed, and each part multiplied by its distance from mid-height of the section in order to find the C.G. of the deck, the shell, and the bottom. The C.G. is usually below mid-height, and through C.G. passes the neutral axis of the section. The area of each part multiplied by the *square* of its distance from the neutral axis, and added together, constitute the moment of inertia. The section is taken in way of openings, but no deductions are made for rivet holes.

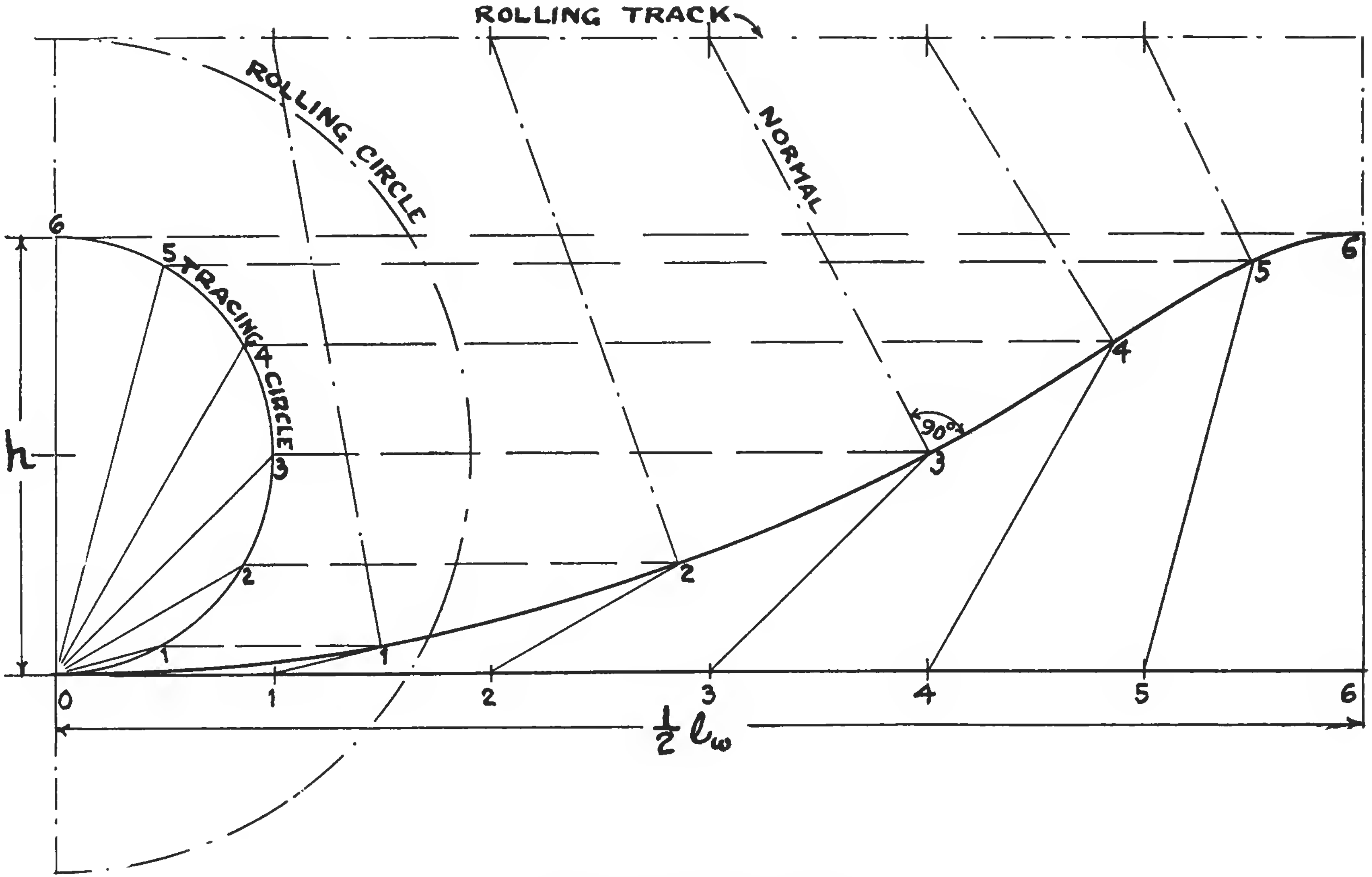


FIG. 41. WAVE PROFILE DIAGRAM.

Instead of measuring distances twice for each part, the first distances from mid-height can be squared for the moment of inertia of each part, but a correction is then needed. The difference of the moments above and below mid-height is divided by the area of the section. The quotient subtracted from mid-height gives the actual position of the horizontal neutral axis. For instance, suppose mid-height of the section is 20 ft., the moment difference = 1560, the area = 23,260, all for one side only, then the neutral axis is 20.0 − (1560 ÷ 23,260) = 19.73 ft. from the base line (the keel). The amount to be deducted from the moment of inertia of the half-section is $23{,}260 \times 0.27^2 = 1700$ (approx.). The British Load Line Committee of 1913 established a minimum thickness of side plating in hundredths of an inch as $0.105L + 17$, and a maximum frame spacing in inches as $0.025L + 17$. For instance, suppose $L = 400$ ft., the side plating should be $(42 + 17) \div 100 = 0.59$ in., and the frame spacing $0.025 \times 400 + 17 = 27$ ins.

The structural material in all cases to be open-hearth steel having a tensile strength of 26 to 32 tons per square inch, and an elongation of 16% or more in a length of 8 ins.

The Load Line Regulations have also given formulas for *transverse strength*, too intricate and too long for insertion here. For both longitudinal and transverse strength there are fixed certain limitations of B and D within which the formulas are valid.

Having calculated the moment of resistance I/y of the midship section, our next step is to determine the maximum bending moment tentatively assumed = W.L./35. First draw two wave profiles (trochoid) of the same length as LWL of the vessel and 1/20 the length in height, Fig. 41, on separate tracing papers. One is to have a crest in the middle, the other a trough, if the longitudinal strength is to be ascertained in both conditions. Next place the crest tracing over the profile drawing of the ship, top of wave at mid-length, and find by trial and error the position of the ship when it displaces a volume of the wave equal to its own weight. Not only must the wave volume equal the weight but the centers must be in the same vertical line. The latter prerequisite makes it sometimes necessary to move the tracing slightly fore and aft.

Our next step is to estimate the weights of the vessel and cargo at each transverse section, and set them up as ordinates together with the volumes. If correctly estimated, the areas of the two curves, and their C.G., must suit each other. The weight estimate is a very tedious

job. To lighten it, Professor Biles[37] proposed for the hull weight a simple outline, Fig. 42, with the *LWL* divided in three parts, and ordinates erected at both ends and at the two intermediate sections. The heights of the ordinates are proportional to hull weight ÷ *LWL*, the percentage for the foremost ordinate = 56.6%, for the middle ones = 119.5%, and for the aftermost = 65.3%, in case the ship is of fine form. For ships of full form the percentages are 59.6%, 117.4%, and 70.6%, in the same order. The outline in Fig. 42 accounts for only the hull weight, weights of the outfit, machinery, fuel, stores and cargo.

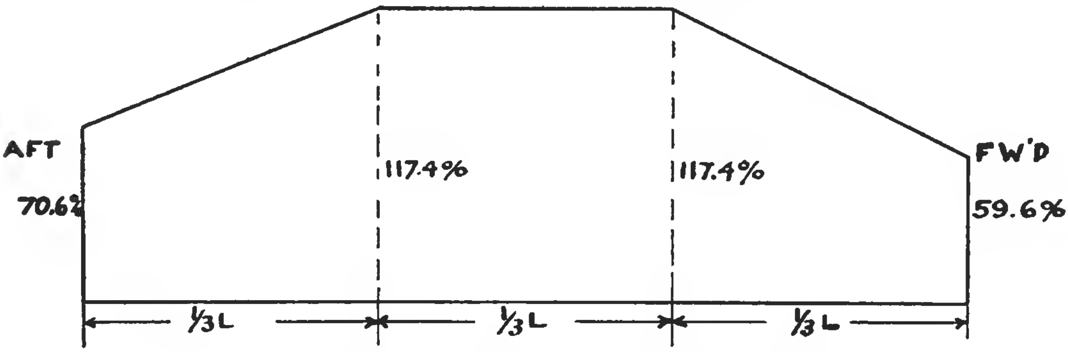


FIG. 42. HULL WEIGHT, APPROXIMATE CURVE.

If the weight of the engine room is to be added, divide the weight by the length of the engine room weight, estimate the C.G. of these weights, and on a separate ordinate through the C.G. add the weight quantity (weight divided by length).

When the weight curve and the displacement curve are complete, the curve of loads is plotted from the differences between the ordinates of the two other curves, below the base line if the weight curve is above the displacement or volume curve, and *vice versa*. The first two curves meet in at least two places, the curve of loads must cross the base line in the same ordinates. The areas of the curve of loads from either end supply the ordinates for the curve of shearing forces, the areas counted positive when the curve of loads is below the base line, and negative when above the base line, in the afterbody. According to the theory of bending, the shearing forces change sign (+ to −) where their curve crosses the base line, hence the areas are counted negative in the forebody when the curve of loads is below the base line, and *vice versa*. Note that the area of the curve of shearing forces below the base line must be exactly equal to its area above the base line. Finally, the areas of the curve of shearing forces from either end furnish the ordinates

of the curve of bending moments, its maximum being above the spot where the curve of shearing forces crosses the base line. It is thus clear why the latter curve must have as much area below the base line as above, otherwise the maximum bending moments would be different when added from the forward end or from the after end.

To repeat: the difference between the *ordinates* of the curve of weights and the curve of displacement or volume gives the ordinates, section for section, of the curve of loads. But the *areas* of the curve of loads supply the *ordinates* of the curve of shearing forces summed up from either end, and the areas of the curve of shearing forces furnish the ordinates of the curve of bending moments. This sequence follows from the theory of bending, for which space cannot be allotted here.

Actually, at sea the heaving, pitching and rolling of a ship greatly increase the bending moments and the stresses, but the excess is usually allowed for by the factor of safety selected. The factor of safety is the ratio between the breaking strength of steel (26 to 32 tons per sq. in.) and the assigned working strength. For instance, a factor of safety of 4.0 means a stress of 6.5 to 8.0 tons per sq. in. Several scientists have tried to make corrections in bending moments from the action of waves, but only that of Mr. W. E. Smith[50] will be mentioned. He suggested that, as the pressure in a trochoidal wave differs from that in still water, the Law of Archimedes is no longer valid in a wave—the buoyancy of the vessel is no longer equal to the weight of the water displaced. He estimated the correction resulting therefrom, since called the *Smith correction*, but it is known that trochoidal waves cannot exist in the ocean (except as bow waves, *see* Appendix) and, as both heaving and settling increase the bending moment, the Smith correction can have only theoretical value. The effect of heaving has been investigated by Mr. C. T. Read,[51] who found that hogging moments were only slightly increased whereas sagging moments were enlarged up to 20% in a fine vessel, and 10% in a full vessel.

The effect of pitching was investigated by Captain A. Kriloff,[52] who found an increase of 80% in the bending moment amidships—momentarily, it is true, but just as destructive, nay, more so than when prolonged. Of course, bending stresses produced at sea by waves are always momentary, although the theory assumes static conditions. A large factor of safety is always necessary. The limiting stress[2] for small ships is usually 6 tons per sq. in., but the calculated stresses for the largest ships often exceed 8 tons in compression, and 10 tons in tension.

Sir Westcott S. Abell[49] has developed a formula for the nominal or calculated stress per square inch ($L = LBP$),

$$p = 5\left(1 + \frac{L}{1000}\right) \tag{60}$$

assuming the standard wave height of $1/20 \times L$. With the writer's proposal, wave height equal to draft of ship, the divisor under L could be altered to $50D$ assuming that the bending moments are proportional to D. This assumption seems to be a fair average, although in some cases it might be more, in others less. The new divisor would make $p = 7.60$ in the case of *Queen Mary* instead of 9.82 from formula (60). For a vessel with a 100-ft. *LBP*, the figures would be 6.67 and 5.50. In view of the allowance for corrosion, grounding, etc., 6.67 tons per sq. in. does not seem to be an unreasonable reduction from *Queen Mary's* 7.60, but the other set of stresses certainly are extremes. Besides, no ocean wave 1000 ft. long can reach a height of 50-ft. in open water.

The maximum tensile stress accepted for *Queen Mary*[49] was 9.32 tons per sq. in. in the upper parts of the hull girder (the promenade deck) assuming a wave height of $L/20$. But with a wave height of $D = 38.83$ ft., this stress would be reduced to 7.21 tons, resulting in a factor of safety of about 5.0, quite sufficient even allowing for only momentary forces. For that matter, the time interval being proportional to the length of the wave, the time factor is more benevolent to the larger vessel—another reason for allotting a higher stress per square inch to the latter.

Scales for Stress Diagrams. It is sometimes difficult to choose suitable scales for the different curves in the stress diagrams. It should be noted that the curve of displacements is not laid off for areas as in a curve of areas but in tons per foot, and a suitable scale is 1/8 in. = 1 ton per foot of length. The same scale is, of course, used for the curve of weights and for the curve of loads, but for the curve of shearing stresses the scale should be 1/8 in. = 100 tons, and for the curve of bending moments a scale of 1/8 in. = 2000 tons might be suitable. These scales apply to a ship of 400 ft. in length; appropriate changes to be made in vessels of different lengths. Of course, all curves have the same length (*LWL* of the ship), except the curve of weights if there are overhangs at the ends.

Stress Calculation. When the maximum bending moments have been developed, and the stress per sq. in. calculated from formula (60), the moment of inertia is obtained from formula (58),

$$I = \frac{My}{p}$$

but in this equation y is still unknown. As far as merchant ships are concerned, the shell thicknesses and all scantlings are tabulated by the Registry Societies from which the moments of inertia of the midship section are estimated. Most of their tables do not go beyond 700 ft., and it becomes necessary to trace the relation between f, the Load Line Committee's factor for the modulus of the mid-section, and L, here the length between perpendiculars. The relation is obtained through the formula $f = CL^n$. The f-value for $L = 600$ is 22.0, and for $L = 300$ is 6.95, from the combination of which n is found $= 1.66$, and $C = 0.00054$, hence

$$f = 0.00054 \times L^{1.66} \tag{61}$$

Checking for $L = 450$ ft., we obtain $f = 13.73$, very close to the committee value $= 13.64$.

Formula (61) gives $f = 51.6$ for *Queen Mary*, whereas the estimated value[49] amounts to 60.9 from an extension of the committee curve of f—always a difficult thing to do by the eye. Taken in conjunction with a wave height of $1/20 \times L$, $f = 51.6$ increases the factor of safety to 5.9, which makes the ship very strong and safe.

The Load Line Committee's equation for I/y, the section modulus, is

$$\frac{I}{y} = fDB \tag{62}$$

from which the modulus is estimated once the principal dimensions of a ship are settled by the designer. Note that the draft D is inserted in formula (62), although $1/20$ of L is still used in estimating the bending moment, instead of D as proposed by the writer.

When a ship is being designed, its maximum bending moment is usually not known at the time that the scantlings are needed. For a nearly rectangular mid-section, it is possible to derive an equation for the side thickness of the plating as well as another for the thicknesses of the decks, inner bottom, and outer bottom. The formula for side thickness t_0 is,

$$t_0 = \frac{fDB}{12H\left(\frac{H}{3} + K_2 \times \frac{B}{2}\right)} \tag{63}$$

H is the molded depth to the strength deck, determined by Freeboard Regulations. The coefficient K_2 is dependent on the number of decks in the ship, as shown in Fig. 43, where the base represents the number of decks, inner bottoms (if any) and outer bottoms, and the ordinates the values of K_2 in formula (63). Then, if t_1 equals the total thickness of the decks, the inner bottom, and the outer bottom, and n equals the number of decks up to and including the strength deck (excluding any deck near the neutral axis) a very simple formula connects t_1 and t_0, namely

$$\frac{t_1}{t_0} = \frac{n + 1.5}{2} \tag{64}$$

For instance, the *Queen Mary* has got six decks plus inner and outer bottoms, hence $t_1 = 4.75t_0$. The actual ratio is 4.60, but most of the upper works are high tensile steel, with consequent reduction of their thicknesses. One deck near N.A. is excluded.

Again, take a single-decker with inner bottom, $t_1/t_0 = 2.25$, the actual ratio for this 265-ft. long ship being 2.10. If the vessel is fitted with a long bridge amidships, the dividend in formula (64) should be $(n + 1.75)$. It is really surprising how accurate this simple formula is.

The distribution of the compound thickness t_1 is regulated by the Registry Societies and based on frame spacing. The frame modulus has been determined by the Load Line Committee, the beam strength depends on the number of pillars or other supports, also prescribed by the Societies.

STEEL WEIGHT FROM STRENGTH. The longitudinal weights of the hull girder represent 36% of the total steel weight, a rule that is sometimes handy for checking purposes. For laps, butts, and rivet heads add 9%, but from this total deduct 16% if the vessel is all-welded.

STRENGTH TESTS ON ACTUAL SHIPS. In 1903 the British Admiralty made some important tests on the small torpedo destroyer *Wolf* (*see* reference 53) and found fairly good agreement between theory and

practice. The neutral axis of the mid-section on which the whole theory rests, was ascertained to be in close proximity to the theoretical position. Measurements were also made of the deflection of the hull under various loads, from which Young's Modulus E was estimated.

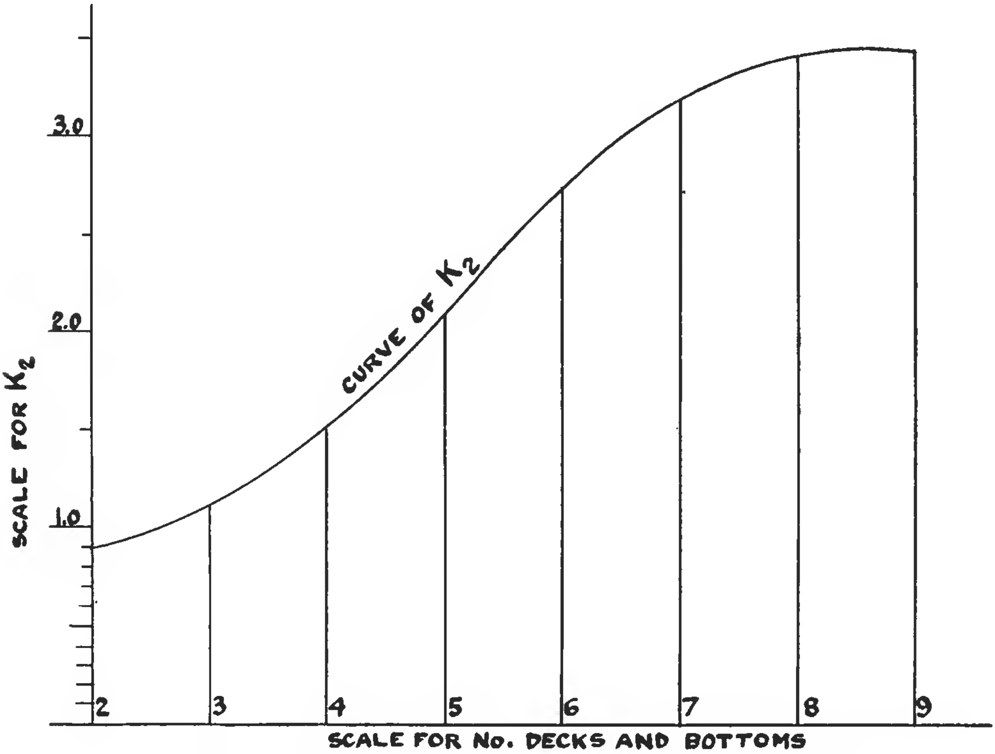


FIG. 43. CURVE OF K_2 IN WEIGHT/STRENGTH ESTIMATES.

DEFLECTION OF SHIPS' HULLS. A ship like a beam is subject to bending from transverse loads. The amount of deviation of the keel or the neutral axis from its position at rest, is termed *deflection*. It has already been mentioned how long Great Lakes vessels hogged 8 ins. when loaded. The theory of flexure introduces a new constant, called Young's Modulus, that shows the relation between stress intensity and strain in elastic materials such as steel and iron. It is denoted by the letter E, and its value is practically the same for tension as for compression. For ship steels $E = 13{,}400$ tons per sq. in.; in the *Wolf* experiments mentioned above, values of E up to 16,000 were obtained in hogging, but for the most severe bending, $E = 9{,}800$ when $p = 6.60$ tons per sq. in., E increasing when p decreased.

In general it is found that ships are not overstrained if the ratio of deflection to length of span $= 1/1100$. The middle span in a ship is

about $0.40L$, hence the deflection referred to L should $= 1/2750$. The theory of deflection states that

$$z = -\frac{Ml^2}{8EI}$$

where z = deflection and l = length of span. Divide both sides by l,

$$\frac{z}{l} = -\frac{Ml}{8EI} = \frac{l}{1100}$$

and if L is inserted,

$$\frac{z}{L} = \frac{l}{2750} = \frac{0.4ML}{8EI}$$

and inserting $E = 13{,}400$,

$$\frac{l}{2750} = \frac{0.05ML}{13{,}400I}, \text{ from which } I = \frac{ML}{95} \tag{65}$$

The negative sign here omitted, means that the deflection is upward when M is negative but is without effect on I. When M is measured in foot-tons and L in feet, I is in square inches $\times$ feet^2.

Example. The *Queen Mary*[49] has got a maximum bending moment of 2,602,500 ft.-tons, her length is 965 ft., her moment of inertia, using formula (65),

$$I = \frac{2{,}602{,}500 \times 965}{95} = 26{,}400{,}000 \text{ sq. in.} \times \text{ft.}^2,$$

but actually it is only 13,960,000, assuming $y = 50$ ft. to promenade deck (strength deck). Hence our initial assumption, $z/L = l/2750$ is evidently too severe, or her middle span larger than $0.40L$.

Addenda. In estimating the weights of steel parts, 1 cu. ft. is usually assumed to equal 489.6 lbs. The steel weight per lineal foot of bars $= 3.40 \times$ sectional area in square inches.

PASSENGER SHIP WEIGHTS

The Rules of the International Load Line Convention for the safety of passengers (if over 12) increase the weights of the ships far above

that of a cargo ship of equal dimensions. Part of the increase comes from greater speed and larger machinery, but subdivision, lifeboats, and passenger accommodation add their part. Of these, lifeboats and accommodation are very difficult items in the weight estimates. The following Table has been compiled from many sources.

TABLE VII

OUTFIT WEIGHTS PER PASSENGER

(Including accommodation and extra lifeboats)

First Class	2.0 tons + 20.0
Tourist Class	1.0 tons + 15.0
Third Class	0.5 tons + 10.0

These weights include the hotel equipment of passenger vessels in ocean or overnight service, and although simple to estimate, are surprisingly accurate. The hotel equipment often comprises more than half of the ship's entire wood, equipment, and outfit weights.

COST OF SHIPS

Some time ago the U. S. Government called for bids on a 9,000 dead-weight-ton cargo vessel, fitted either with steam turbines or Diesel engine machinery. The bids from seven shipyards varied 175% for the steamship, and 188.5% for the Diesel ship. Nothing can more fully show the difficulties of cost estimating, for neither the materials nor the labor costs could vary more than a few percent at the different yards, and probably all added the same percent for profit. If measured per D.W. ton, the bids ranged between $212.00 and $370.00 for the steamer, and still more for the Diesel ship.

The cost estimates include the building or first cost, and the operation costs, but shipyard bids are concerned only with the former. The Naval Architect, however, can largely influence the latter by choosing the optimum dimensions and the most suitable speed-length ratio, as proposed in the previous text.

The most profitable size, dimensions, and speed of cargo ships have been investigated by many experienced men, the earliest estimates by Mr. John Anderson,[54] and the most scientific by Mr. John Tutin,[55] D.Sc. In no case were the optimum dimensions or the best speed-length ratio considered, nor the fact that the cargo goes to the fastest ship as a rule. Hence all investigators found that the slow ship was

the profitable ship—and no mistake! Or as a prominent shipowner told the writer, "What do we gain by low fuel consumption if our vessels are too slow to get cargo?"

Ships can be classed in at least seven divisions: 1) cargo ships, 2) tankers and colliers, 3) passenger vessels, 4) yachts, 5) warships, 6) ships for special purposes, 7) pirates. Although these vessels cover wide variations in types and sizes, their cost estimates can be worked along fairly similar lines if brought together in two main groups: merchant ships and yachts, and warships. The separate cost accounts might then be named as follows,

Merchant Ships and Yachts	*Warships*
Estimating	Estimating
Drafting	Drafting
Model work	Model work
Mold loft work	Mold loft work
Steel hull	Steel hull
Hull engineering	Hull engineering
Wood and outfit	Hull outfit
Hotel equipment	Armor backing
Pumping and drainage	Fitting armor in place
Plumbing and ventilation	Turret machinery
Propulsive machinery	Gun placements
Auxiliary machinery	Fitting guns in place
Launching work	Ammunition hoists
Painting, etc.	Military masts
Trial trips	Fitting in place other items supplied by the government
Overhead	Pumping and drainage
Profit	Plumbing and ventilation
	Propulsive machinery
	Auxiliary machinery
	Launching work
	Painting, etc.
	Trial trips
	Overhead
	Profit

Any of these main accounts can be subdivided as far as the Chief Estimator wants it. In the overhead expenses figure, for instance, Office Workers, Foremen, Timekeepers, Maintenance, Bonds, Freight, and Insurance. The overhead is always estimated as a percentage on all accounts ahead of it, and the profit, of course, as a percentage of the total cost.

The basis of all cost estimates is price per pound which, when multiplied by the weight of the item, gives the cost. Most prices are constantly changing either up or down, hence the cost of ships must clearly change with the prices. Fig. 44 shows how prices of a 7,500-ton cargo ship[56] have fluctuated during the years 1898–1912. The writer in

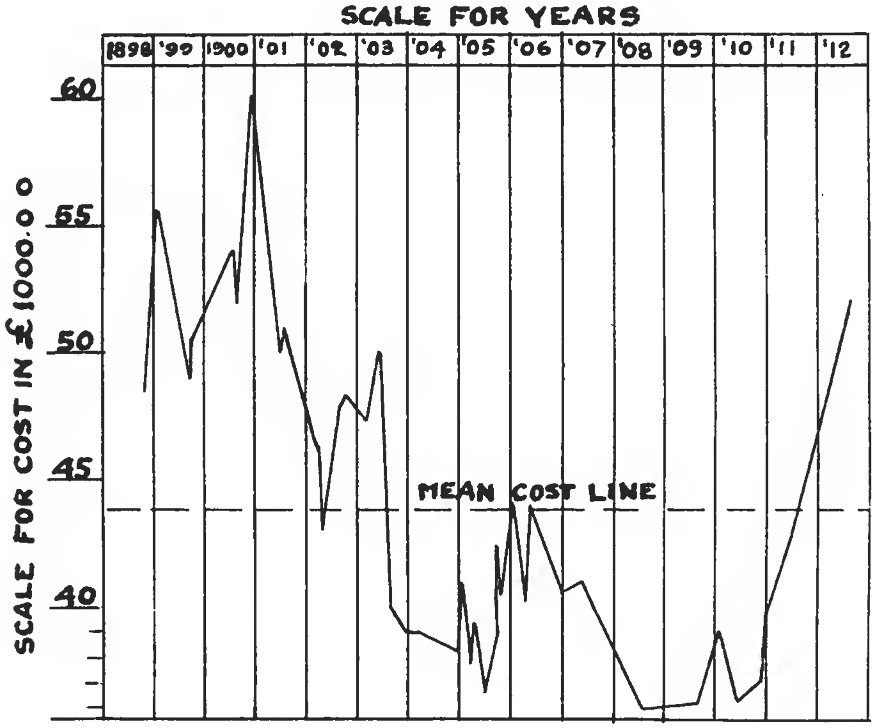


FIG. 44. TYPICAL SHIPS' COSTS, 1898–1912.

a paper before the Institution of Engineers in Scotland proved that freight rates fluctuate in much the same way. Price statistics have been gathered into price indexes for many years. The first one was Sauerbeck's Index begun in 1818 and published in the British periodicals *Fairplay*, *Statist*, and others. In the United States, several indexes are now published by the Government and private firms. As a rule the fluctuations are so strong that no fair curve can be made to conform to their antics. But by taking 20-year means Mr. Axel F. Enstrom, Ph.Dr., President of the Swedish Board of Trade, succeeded in developing fair, undulating curves of the Index that could be extended to predict the economic future.[57] By the help of the French wheat statistics,[58] Mr. Enstrom was able to reach backwards until 1250 A.D. and to prove almost complete agreement between the Index and the

sun spots! His Index curve shows that the future fluctuations will be less violent than those from 1850 to 1910.

A wide-awake cost estimator cannot thus base his costs on the present prices only, but must look into the future, since several years may intervene between his estimates and the delivery of the ship. He might especially keep watch on the index high spots which Mr. Enstrom's curve indicates in 1944, 1953, and 1963.

Although nothing can take the place of a detailed cost estimate, a simplified formula of costs is very useful for checking and for quick estimates. The costs of materials in a ship are always proportional to their weights, but labor costs are dependent on both weights and surfaces. Labor costs in painting, insulation, etc., are estimated by surface, hence it is erroneous to estimate labor as so much per pound of the structure. If the ship is taken as a whole, her weight is the light displacement without any stores, fuel, or water aboard. Her surface is proportional to the two-thirds power of the load displacement, but varies somewhat with the type of the ship. A tanker, with its closely-spaced bulkheads, possesses a large surface, but most of the inside is left unpainted, whereas a passenger vessel has got plenty of surfaces all painted. The cost of machinery varies nearly as the horsepower, which is dependent on the speed-length ratio. The simplest formula would then be

$$\text{cost of ship} = k^2(C_1 \times W_{\text{light}} + C_2 \times W_{\text{load}}^{2/3}) \qquad (66)$$

k = speed-length ratio, in knots and feet
W_{light} = light displacement in tons
W_{load} = load displacement in tons
C_1 and C_2 = constants depending on type of ship

It is not possible to give reliable values of C_1 and C_2, but the estimator can easily obtain them by comparing two ships of the same type built in the same year, and correcting these values for Index variations.

Another, more accurate formula takes account of the percentages of direct labor, of materials, and of general expense (overhead) for the main cost items of the ship. As an instance, a cargo ship of 10,400 D.W. tons has been taken, and percentages worked out in Table VIII.

It is usual to add 7% the hull steel weight for scrap.

From these basic percentages cost *formulas* were developed for the main cost items, as follows, based on the 1928 Index:

$$\left.\begin{array}{l} \text{cost of steel hull, weight} \times \left(107.6 + \dfrac{36100}{LBP}\right) \\ \text{cost of wood and outfit, weight} \times 241.2 + (\text{weight})^{2/3} \times 1180 \\ \text{cost of machinery, weight} \times 425 + (\text{weight})^{2/3} \times 3000 \end{array}\right\} \quad (67)$$

Weights are measured in tons at 2240 lbs.
Profit *not* included in formulas.

When using the Index values for changing the constants, material costs for the three main items must be altered separately, and labor and expense can be taken together. For instance, in 1932 the steel hull cost was 74% of the constants in formula (67), the wood and outfit cost 76%, the machinery cost 79%, but the labor and expense cost was only 90% of the 1928 values. Costs have since risen tremendously. For instance, the cost of the 9,000 D.W.-ton cargo vessel mentioned at the beginning of this section, estimated from formula (67) come out to $1,575,000.

The lowest bid was for $1,857,000, the highest $3,240,000. The contract was signed on a bid of $2,029,000.

COST OF DUPLICATE SHIPS. Sometimes two or more identical vessels are ordered from the same shipyard, in the hope that costs might be reduced, and time saved. While such hopes are generally fulfilled, the loss of experience from slightly different vessels is a serious drawback that may outweigh any advantages as regards economy. There is little or no saving in the cost of materials. Most of it is derived from ordering, drafting, and lower freight rates.

TABLE VIII

COST OF CARGO SHIPS

Percentage Cost of 10,400 D.W. Tons Cargo Ship fitted with Geared Turbines and Watertube Boilers. Length of Ship 465 ft. Speed 14 Knots, Shaft Horsepower 6,000, Speed-Length Ratio 0.65.

Item	*Direct Labor, %*	*Materials, %*	*General Expense*	*Totals, %*
Hull, Steel	14.02	18.00	8.50	40.52
Wood and Outfit	2.72	7.76	2.06	12.54
Machinery	6.53	23.40	6.53	36.46
W.T. Boilers	2.93	4.62	2.93	10.48
Totals %	26.20	53.78	20.02	100.00

Note.—Hull Steel Cost includes Plans, Models,Fees, Staging, Shoring, Cement, Paints, etc.—Wood and Outfit Cost includes Plans, Paint, etc. Complete Machinery Cost includes Plans, Fees, Testings, Trial Trips.

Mr. Homer Ferguson[59] stated that if two ships were built at a time, the savings per ship would be 6.25%, and if five ships were built, the savings would reach 11%. The ships quoted were 450 ft. long, speed 18 knots, cost $4,000,000.—If the given percentages were correct, the cost of a single ship would be increased by 6.75% as compared to two ships at a time, and by 12.25% if five ships (*see below*).

The number of ships affected by duplicating is always one less than the total number to be built. The ratio of ships affected is $\frac{N-1}{N}$ and not N, because if $N = 1$, no ship is affected and the ratio too $= 0$.—On the other hand, if the cost of N duplicate ships is known, the increase in the cost of building a single ship $= \frac{S}{1-S}$, where $S =$ savings, as given in Fig. 45. From the values of savings, and the reasoning given above, the general equation of savings from duplicate ships is found to be

$$S = 14.30\left(\frac{N-1}{N}\right)^{1.20} \tag{68}$$

From this equation the curves in Fig. 45 have been plotted.

In case the ships are not 450 ft. long, a correction must be *added* to the savings of all lengths below 450 ft., and deducted for all lengths *above* 450 ft. For instance, if a proposal were for five ships of 300 ft. in length, the saving per ship would be 11.0% for the 450-ft. vessel plus 0.65 for the shorter ship, total savings 11.65%. The lower curve in Fig. 45 crosses the base line at a length of 450 ft.; ordinates above the base line are to be added, those below to be subtracted from the percentage of the given number of 450-ft. ships.

The percentages shown are practically independent of the type of the ship, with the exception of tankers, where the savings might be up to 10% less on account of the more exacting work.

Costs of Welded Ships. It has been mentioned that the weight of the welded steel hull is 16% less than the riveted hull of a similar ship, but very little has been given out about the costs. *Marine Engineering*, October 1941, publishes some data of Swedish welded tankers that can be used as starting points in estimating costs. The displacement of the welded hull is 6% less than that of the riveted hull. This would check with the 16% less weight of steel, if the hull steel weight

amounted to 37.5% of the load displacement—far too much, since the D.W. reaches 75% of the displacement in these big tankers of 16,000 tons D.W. This would reduce the displacement by only 3.5% instead of 6%, and the wetted surface by 2.2%.

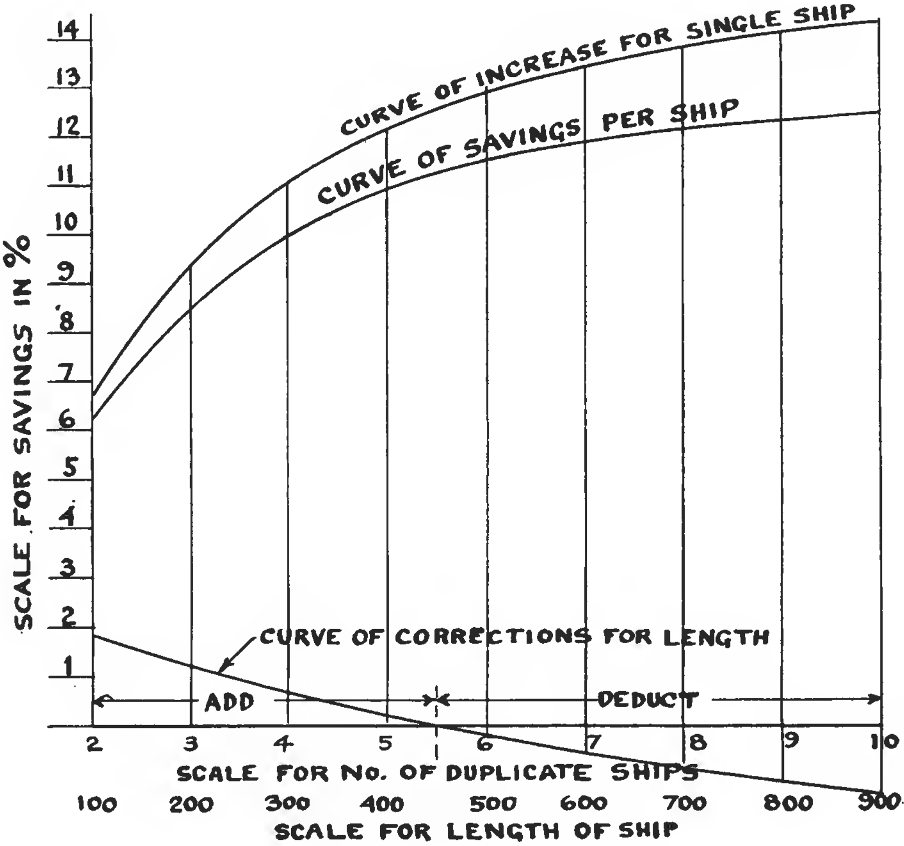


FIG. 45. DUPLICATE SHIPS' COSTS IN PERCENTAGE.

According to the same source, the shaft horsepower would be 11–12% less for the riveted ship—another mistake. The elimination of laps, butts, and rivet heads saves 4% of the skin resistance, the reduced displacement counts for 2.3% less pressure resistance. But the latter is only about one-half as high as the skin resistance, hence the saving in shaft horsepower is 5.4%, or less than half of the Swedish estimate. Assuming that the cost of welding per pound is the same as the riveting cost, the welded ship will save 16% of the steel hull cost, and 5.4% of the machinery cost. For the 9,000 D.W.-ton ship, the cost would be reduced by 9.25%.

UNIT WEIGHTS OF MATERIALS

1. Liquids. Unit: Cubic Foot, 1 Cubic Foot = 7.48 Gallons.

Name	*Weight in Pounds*	*Name*	*Weight in Pounds*
Water, fresh	62.4	Gasoline	45.5
Water, salt, in ocean	64.0	Petroleum (kerosene)	50.0
Oil, boiler fuel	56.0	Lubricating oil	57.5
Oil, Diesel fuel	54.0	Aviation fuel	45.2

2. Solid Bodies. Unit: Cubic Foot.

Aluminum	162.0	*Dry Woods:*	
Aluminum bronze	480.0	Ash	50.0
Brass, naval	525.0	Butternut	35.0
Gun metal	550.0	Cedar, white	25.0
Bricks and mortar	113.0	Cedar, red	22.0
Copper	550.0	Mahogany, Honduras	35.0
Coal (43 cu. ft. = 1 ton)	52.0	Mahogany, Santo Domingo	54.0
Cement and sand	130.0	Oak, white	50.0
Cork, compressed	15.0	Oak, live	67.0
Glass	165.0	Pine, white (mean)	30.0
Iron, wrought	480.0	Pine, yellow (mean)	45.0
Iron, cast	450.0	Spruce	30.0
Lead	710.0	Teak	56.0
Linoleum	75.5	Hickory (mean)	46.8
Nickel, cast	521.0	Elm (mean)	43.0
Phosphor bronze	549.0		
Steel	489.6		
Zinc, cast	435.0		

3. Insulation, Tiling, Paints. Unit: Square Foot.

Name	*Thickness in Inches*	*Weight in Pounds*
Masonite	7/16	0.65
Insulite	7/16	0.61
Maftex	7/16	0.75
Celotex	7/16	0.60
Firefelt covered by No. 16 gauge metal	1.5	5.70
Cork slabs covered by No. 16 gauge metal	1.5	5.90
Tiling, ceramic, including cement 1/16″	7/16	7.30
Tiling, rubber, including cement 1/16″	1/2	5.10
Painting, 3 coats on steel		0.30
Painting, 3 coats on wood		0.21

BREAKING STRENGTH OF WIRE ROPE

(Material: Crucible Steel)

Circumference in Inches	*Diameter in Inches*	*Strength in Pounds*	*Circumference in Inches*	*Diameter in Inches*	*Strength in Pounds*
1	5/16	4000	2 1/4	3/4	23,000
1 1/8	3/8	5700	2 1/2	13/16	27,000
1 1/4	7/16	7800	2 3/4	7/8	31,000
1 1/2	1/2	10,000	3	1	39,000
1 3/4	9/16	13,000	3 1/4	1 1/16	45,000
2	5/8	16,000			

Appendices

I

Theory of Trochoidal Wave

The motion of ocean waves, seemingly so simple, has been investigated by many learned men, but its theory is as far from final solution as when F. J. von Gerstner in 1802 read a paper before the Royal Bohemian Scientific Society. The paper was discovered by Professor Rankine in 1862, but both authors made some preliminary assumptions that were open to serious objections. These objections were eliminated by Dr. Joseph Woolley in a most scientific paper read before the Institution of Naval Architects (London) in the late 70's.

Dr. Woolley's paper was assiduously studied by the writer during his senior years in college but he never succeeded in mastering it, because Dr. Woolley had evidently left some big gaps in his theory—gaps that seemingly could be bridged only by himself, for neither the *Encyclopedia Eritannica* nor any other work has ever mentioned Dr. Woolley's paper after it was read. Yet, his conclusions were strictly logical, and his equations of motion confirmed the findings of earlier investigators in that the only possible paths of the water particles were circular in trochoidal waves. As a circular or rotational motion cannot be given to the particles by any natural means such as wind and gravity, the conclusion must be reached that ocean waves are not trochoidal in outline. However, both in length, speed, and period, the ocean waves agree almost exactly with the theory of trochoidal waves, which fact is of greatest importance for several investigations of naval architectural problems such as the rolling of ships, the strength of ships, but especially for the pressure resistance of ships.

How far does actual ocean wave conform with the trochoidal theory? —A casual observer on a big pier jutting out in the ocean as, for instance, at Atlantic City, New Jersey, sees small objects bobbing up and down as the waves pass inshore. Looking down from the outer end of the pier, he also sees the objects moving back and forth. Hence it is clear that the water particles must move in circular or rectangular paths parallel to the direction of the waves. Any object that possesses

mass can move only in a curved path, and that throws out the rectangular path. There remains only the circular and the elliptical paths, if we do not concede the possibility of a path looped like the figure 8, either standing or lying. In any case the orbital velocity in the path might be assumed to follow *Kepler's Second Law*, so that the velocity is in inverse proportion to the focal radius of the path. Only in a circular path can the orbital velocity be constant, which is one of the assumptions of the trochoidal wave theory.

In a paper written in 1899 and rejected by the Society of Naval Architects (New York), this writer proved that the orbital velocity at the surface of the bow waves = $V \sin a$, the divergent velocity of the water particles thrown aside by the bow of the ship. In order to assist the formation of the waves, the vertical sections of the bow should be slightly wedge-shaped—a form that has persisted for a hundred years to this day.

Although this theory takes account of the divergent waves, it cannot explain the existence of the more important transverse waves that have been shown to agree exactly with the trochoidal wave theory as regards length, speed, and period. The only exception seems to be in some of Taylor's experiments, where the bow waves flooded the deck near the stem of the model, disturbing the natural arrangement of the waves, both the divergent and the transverse waves. But whereas the divergent waves, once formed, roll away from the side of the ship, the transverse waves are in contact with the vessel for its full length, mightily influencing its wave resistance. There is also another great difference between the two systems of ships' waves, in that all the particles of the transverse waves are supposed to reach the crest at the same instant, while those of the divergent waves reach the crest at successive moments. In the former case, the crest line stands square to the course of the ship, in the latter case the crest line forms a helice together with the trough line and intermediate spots. The helical crest line shows conclusively that the particles revolve in vertical planes square to the direction of the waves.

The helical crest line evidently gives rise to the successive crests that we call waves, and curiously enough these crests seem to keep exact time with the transverse waves according to Froude. But the Swedish engineer, V. W. Ekman, showed in a pamphlet *On Stationary Waves in Running Water* in 1906, that there exists a difference in the phase between the transverse and the divergent waves at the outer common

boundary of the two series. His explanation of the formation of the two wave systems, as given by the *Encyclopedia Britannica*, is not very convincing, and leaves the formation as much a mystery as ever.

WAVE PROFILES. There are several mathematical curves that can serve as wave profiles on paper. The simplest one is called the Sinusoidal Wave Curve as developed in the Theory of Sounds. It is easily plotted with the help of a Table of Sines, starting in the axis where $\sin a = 0$. This happens first when $a = 0$, but subsequently when $a = 180°$ and $360°$. These spots on the axis are termed *nodes;* and translated to water waves, the axis itself represents the undisturbed surface of water. When $a = 90°$, $\sin a = 1.0$, and when $a = 270°$, $\sin a = -1.0$ (negative), which interpreted means that the curve is proceeding in a series of humps and hollows, the hollows exactly equal to the humps if reversed. The mathematical expression for this curve is

$$Y = r \times \sin \frac{2\pi}{l_w} \times x \tag{1}$$

where y = any ordinate of the curve
x = any abscissa
l_w = length of the wave
r = half-height of wave
π = 3.14

If the wave outline is supposed to travel along with the speed V in feet per second, its *period* or the time in seconds that it takes to travel its own length, equals $\frac{V}{l_w}$, and its mathematical expression becomes

$$Y = r \times \sin \frac{2\pi}{l_w} (x - Vt) \tag{2}$$

where t signifies time in seconds. By choosing t as fractions of the period V/l_w ($t = V/2l_w$, etc.), the position of the crests of the wave from any starting point can be fixed.

This explanation is merely given as an introduction to the more intricate theory of trochoidal waves. The movements of the water particles in a sinusoidal wave is simply up and down; but, as mentioned, the movement of the particles in an ocean wave is both up and down

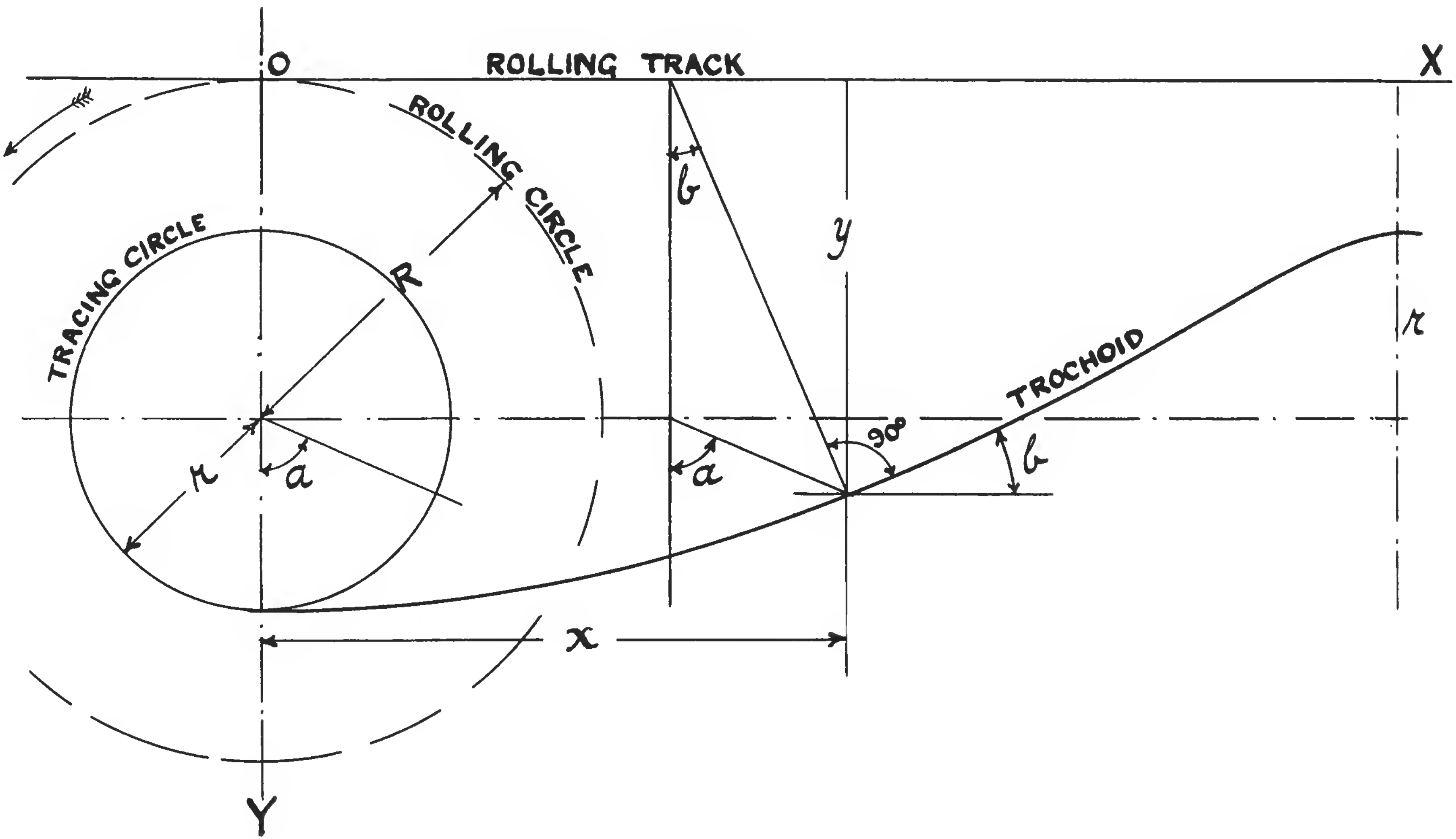


FIG. 1. CONSTRUCTION OF A TROCHOID.

and sideways. Hence the profile of the ocean waves cannot be sinusoidal, although the long swell met in a calm after a big storm, approximates the outlines given by formulas (1) and (2). The basic form of the trochoidal wave profile is the inverted cycloid formed by a fixed point on a circular arc as it rolls along a straight line *above* it. This curve represents the wave at its breaking point, its length l_w only 3.14 times its height $2r$. The mathematical equations of the cycloid are,

$$\left.\begin{aligned} x &= r(a - \sin a) \\ y &= r(l - \cos a) \end{aligned}\right\} \tag{3}$$

The angle a is here measured in *radians* of which 3.14 radians = 180°. Thus l radian = 57° 18′. For instance, assume a = 45° or 0.785 radians, $\sin a = 0.707$, $\cos a = 0.707$, hence

$$x = r(0.785 - 0.707) = 0.078r$$

$$y = r(l - 0.707) \qquad = 0.293r$$

The abscissa axis is the straight line on which the circle rolls.

If the fixed point is located *inside* the rolling circle, the curve described by the point is termed an elongated cycloid or a *trochoid* for short. Fig. 41, page 170, shows how a trochoid is drawn on paper once the length and the height are known, but in order to understand its geometry, both the rolling circle and the tracing circle must be shown as in Fig. 1. The radius R of the rolling circle multiplied by π, equals half the length of the wave, and the radius r of the tracing circle equals half the height of the wave. The mathematical equations of the trochoid are,

$$\left.\begin{aligned} x &= Ra + r \sin a \\ y &= R + r \cos a \end{aligned}\right\} \tag{4}$$

The term Ra equals the distance traveled by the rolling circle after it has revolved through an angle a, and this distance is generally either longer or shorter than x, the ordinate of the fixed point. Only in the exact center of the wave hollow, where $x = 0$ and $y = R + r$, and in the center of the wave crest, where $x = 0.5\ l_w$ and $y = R - r$, is $Ra = x$. Note that both $\sin a$ and $\cos a$ become negative at certain angles, as noted above, which makes the trochoid quite unlike in form to the sinusoidal wave profile—with shorter and steeper crests, and longer and flatter hollows.

Both the cycloid and the trochoid have a common peculiarity, in that the normal to the curve at any point P passes through the spot A, where the rolling circle touches the straight line track at the same instant. The slope of the curve is thus equal to the angle between AP, the normal, and the vertical ordinate at the spot A. Since in Fig. 1 y is counted positive downwards from the track or X-axis OX, contrary to usual analytical practice, $\tan b = -dy/dx$ when the slope of the curve is upwards toward the right. If the two equations in formula (4) are differentiated,

$$-\frac{dy}{dx} = \frac{0 - r \sin a \times da}{R \times da + r \cos a \times da}(-1) = \frac{r \sin a}{R + r \cos a} \tag{5}$$

from which equation the slope can be obtained at any point. The maximum slope is found by differentiating formula (5) according to the rules of the calculus of maxima. It is at the same time the inflexion point of the wave profile, and is nearer to the crest, the shorter and steeper the wave. In the cycloid, the inflexion point is at the very crest: in the trochoid it is quickest found by direct construction of the curve as in Fig. 41, page 170.

The normal AP to the curve in Fig. 1 is also the instantaneous radius of the point P which means that P is revolving momentarily around the spot A as a center.

In order that the trochoid may represent a possible wave motion, the hydrodynamical principles of equilibrium, continuity, and internal pressure must be satisfied. By further developing the trochoidal wave theory along lines laid down by Gerstner, Rankine, and Froude, it can be proved that these principles are fulfilled, and that the trochoids at different depths represent lines of equal pressure. Although the radius R of the rolling circle remains the same (otherwise the crests in the lower strata could not keep step with the surface crests), the radii of the tracing circles grow smaller and smaller with the depth, and become practically 0 at a depth equal to the length of the wave.

The speed of the wave $= \sqrt{\frac{gl_w}{2\pi}}$ in feet per second, equivalent to

$$V_w = \sqrt{1.80\, l_w} \text{ in knots per hour} \tag{6}$$

The period of the wave is the time it takes to pass from crest to crest, or

$$T = \frac{l_w}{V_w} = \sqrt{\frac{l_w}{5.11}} \text{ in seconds} \tag{7}$$

A peculiarity of the trochoid wave motion is that the orbit of any particle (assumed to be a circle) is raised above its still water level a distance

$$h = \frac{r^2}{2R} \tag{8}$$

For example, if a standard wave length is assumed to be equal to twenty times its height, $l_w = 2\pi R = 20 \times 2R = 40r$, hence $r = \frac{2\pi}{40}R$, and $\frac{r^2}{2R} = \frac{(0.157\ R)^2}{2R} = 0.0123\ R = 0.0785\ r$. If the wave were 10 ft. high, the centers of its surface orbits would be raised $0.0785 \times 5 = 0.393$ ft. = 4.70 ins.

The energy in a wave according to the trochoidal theory, is half kinetic and half potential. In salt water weighing 64.0 lbs. per cu. ft., the total energy, in foot-pounds per foot of breadth,

$$E = \frac{1}{8} \times 64.0 \times l_w \times \frac{r^2}{4} = 2l_w \times r^2 \tag{9}$$

Assuming a wave 10 ft. high by 200 ft. long, its energy $E = 2 \times 200 \times 5^2 = 10{,}000$ ft.-lbs. = 18.2 horsepower.

In creating a wave system, the kinetic energy only needs constant renewal, but as there are two wave systems to each side of the ship, the values in formula (9) should be multiplied by 2, making the wave resistance $HP = 36.4$ at a speed of 19 knots, formula (6). This is clearly erroneous, hence wave energy is no measure of wave resistance of ships.

II

Capacities of Merchant Ships

Capacities of merchant ships should be considered from two main standpoints, profit-earning, and cargo delivery. The profit-earning qualities of ships have been treated by many eminent investigators (*see* page 179), but none of them made allowance for the influence of the favorable speed length ratios 0.63, 0.72, 0.85, and 1.11. In fact, since the optimum form, proportions, and dimensions of ships were unknown at the time, this influence could not have been taken into account anyway. The conclusions of the earlier investigators that the slow-speed ship was the profit-earning ship, have, therefore, not been confirmed by actual practice.

Perhaps the most drastic example in the case comprised the 450 odd U. S. Shipping Board vessels, each of 10,000 tons D.W., rotting in American harbors for twenty years. Not one could be made to earn its keep in peacetime.

As regards the cargo delivery quantities, pure dead weight does not give a true measure of this important *phase* of the shipping business, except for ships of the same speed running on the same route or on the same length of voyage. Speed and distance covered thus become necessary considerations in an analysis of successful merchant ships. It is not the dead weight but the cargo delivered that marks such a vessel as outstanding.

Profit-earning is, of course, the only incentive to build ships in peacetime, whereas cargo delivery becomes the only criterion in wartime when a maximum amount of cargo must be transported in a given period, and expenses do not count at all. Cargo weight is the difference between the dead weight and the weight of fuel, water, stores, crew and passengers, etc., combined. The faster the ship, and the longer the trip, the larger becomes the combined weight, and the less the cargo delivered. For a comparison of different vessels, speed and distance must receive attention but this is not enough—there must be another common denominator as well. First cost or displacement can

be used for this purpose, but as first cost is very elusive as well as varying, and as displacement is more or less fixed, the latter is better for the purpose.

Next come the speed-length ratios. Of the four mentioned above, the lowest, 0.63, can be discarded as it would mean a speed of only 12.6 knots for a 400 ft. vessel, a speed that recent practice has found too low. Of the remaining three, the problem is then to find which one is the most favorable for the shipowner in peace, or for the military objectives in wartime. The latter problem presenting the least difficulties, our analysis will take it first.

Coastwise Vessels. Up to a draft of 14 ft., coasters can be designed according to the Law of Similitude with similar proportions within the same speed-length ratio. In this case the hull weights favor the larger ship, and the machinery weights the smaller, hence the ratio dead weight/displacement remains almost constant. This fact is proved in the *Shipbuilding Cyclopedia* by Mr. James L. Bates, at present Technical Director with the U. S. Maritime Commission, who used an immense mass of ship data for his Diagram. His vessels were all steam-driven, their hulls riveted, but as this analysis considers Diesel-engined ships, all-welded, the Bates ratios have been increased by 11% to agree with present ratios.

But, although the ratio dead weight/displacement thus remains constant within the same speed-length ratio, the cargo weight per ton of displacement is steadily decreasing as the speed is increasing, because the weights of fuel, water, etc., are proportional to the shaft horsepower and the time spent at sea. Fuel for Diesel engines may be taken at 0.45 lbs. per SHP per hour for coasters, and the consumption for T days,

$$= \frac{0.45 \times SHP \times 24T}{2240} \text{ (tons)} \qquad (1)$$

To this amount, 15% is added for emergencies, and finally 33% of the total fuel weight to include lubricating oil, water, stores, crew weights.

In estimating shaft horsepower from ship resistance, the difficulty is met that Diesel engines are usually running at high revolutions causing high slips and low propeller efficiencies, seldom over 45% for coasters of slow and medium speeds. From a final design, resistance,

wake, thrust, and screw efficiency can, of course, be calculated but the results should always be checked with recent practice in similar vessels.

Since coasters occasionally make long voyages, the total fuel weight that a vessel can take on is always more than is needed on short coasting trips, but in this estimate only enough fuel for the round trip is assumed in each case.

The fuel weight and the dead weight known, the cargo weight is obtained by deducting the fuel weight from the dead weight. Since the ratio dead weight/displacement is constant, it seems best to use ratios for all three weights according to the formula, cargo weight/displacement = dead weight/displacement − fuel weight/displacement. Finally, the cargo weight multiplied by the number of round trips per year gives the total cargo delivered in that time.

Cargo Delivery. This aspect of the shipping business is more easily analyzed than profit earning, and will be considered first. As already mentioned, when comparing ships at different speed-length ratios, their displacement should be the same. For our coasters, a displacement of 1,800 tons is assumed, which furnishes the following particulars:

$V/\sqrt{L}$	L	V	*Cargo Weight*	*Fuel Weight*	*Dead Weight*
0.72	180.5	9.67	1,380	41.0	1,421
0.85	197.5	12.0	1,235	65.0	1,300
1.11	211.5	16.2	1,061	126.0	1,187

Certain investigators have used the amount of cargo delivered per voyage as a basis for comparison. These particulars show that the fastest ship can only deliver 77% of the cargo that the slowest ship can transport. The former ship costs much more, her fuel expenses are nearly twice as large, hence no wonder that the slow-speed ship was considered the most profitable one. Evidently the amount of cargo delivered in a certain period should be used as basis.

Usually the period is a year, but allowing for overhauls, Sundays in harbor, and other delays, 330 days are counted as a year. The number of days at sea per year is thus equal to the length of the voyage in nautical miles divided by 24V, the mileage in 24 hours. But to this time must be added the time spent in harbors, clearly depending on the amount and on the class of the cargo. A long ship can be given more hatchways, hence length plays a rôle, too. Carl E.

Petersen[41] proposes cargo dead weight divided by ship's length and by a constant = 2.5 as a standard for time in harbors, but this should be reduced with modern loading appliances by using a constant = 3.0. For instance, the slowest ship above would spend in harbor,

$$\frac{1{,}380}{180.5 \times 3.0} = 2.5 \text{ days},$$

which seems to be reasonable.

The number of round trips per year equals 330 divided by the total time spent at sea and in harbors. However, while full cargo can generally be counted on for one trip, the return trip might sometimes furnish much less cargo, and half of full weight seems to be a fair average. The time in harbor is also cut in two.

Assume now a *round trip* of 2,000 miles for the coasters, and the vessels would fare as follows:

$V/\sqrt{L}$	*Time Spent at Sea*	*Time Spent in Harbor*	*Total Time*	*Round Trips per Year*	*Cargo Delivered*
0.72	8.60	3.75	12.35	26.70	47,500
0.85	7.00	3.10	10.10	32.70	51,750
1.11	5.15	2.55	7.70	43.00	58,000

The high-speed ship thus delivers 22.0% more cargo than the slow-speed vessel. The fallacy of building slow-speed vessels during war-time is thus demonstrated. No allowance is here made for bad weather at sea that should slow down all three ships proportionately.

In case the vessels are given a beam = $L^{2/3} \times$ constant, an investigation shows that the fastest ship delivers the most cargo. The case might be different in peace time when profit-earning is more important. It will be treated later.

Ocean-going Ships. Such vessels should always be given a parallel middle body of certain length which precludes their being in accord with the Law of Similitude. Curiously enough, the resistances of the ships are still nearly proportional to the displacements, but the proportion constant evidently increases slightly with the speed-length ratio. In the case of coasters, this increase is about balanced by the finer runs of the faster ships resulting in less thrust increase and higher propeller efficiency. It remains to be seen if this holds good for ocean-going ships, too.

In order to obtain the cargo weight, as before, the dead weight and the fuel weight must be estimated in each case. The ratio of dead weight/displacement is as follows for each favorable speed-length ratio, somewhat higher than for coasters:

Speed-length ratio	0.63	0.72	0.85	1.11
Dead-weight ratio	0.792	0.740	0.675	0.540

As already mentioned, the lowest speed-length ratio should not be considered except for the designs of very large cargo ships of 20,000 tons D.W. or more.

In order to compare ocean ships of different speed-length ratios, as before, a common displacement should be used, and one of 18,000 tons is assumed here—large enough to accentuate all *pros* and *cons*. The vessels will have the following particulars:

$V/\sqrt{L}$	L	V	*Cargo Weight*	*Fuel Weight*	*Dead Weight*
0.72	469.0	15.6	11,120	2,200	13,320
0.85	510.0	19.3	8,630	3,520	12,150
1.11	550.0	26.1	2,900	6,650	9,550

From this table, the bad effect of stability and seagoing requirements on cargo capacity is evident, for the increase in speed between the slowest and the fastest ship can never equalize the tremendous reduction in cargo weight. In order to comply with the Law of Similitude, the fastest vessel should have a beam of over 102 ft., and a draft of over 35 ft. While such a great beam is not excessive in battleships, the resultant metacentric height would make a merchant ship very uncomfortable in a seaway. But what a cargo she could carry! And, although her engine power would be excessive at a speed of 26.1 knots, her resistance per ton of displacement would be the smallest and her cargo the largest possible, on a displacement of 31,000 tons, as against the 18,000 tons in the table. It may be argued that the speed is too high for a merchant ship, but with their speeds now reaching 20 knots, who can tell? Notice from the Table of Coasters above that the fastest ship delivers the most cargo per year.

It all goes to show the tremendous importance of giving cargo ships the greatest beam and draft commensurate with harbor depths of their trade routes.

Assuming a *round trip* of 14,500 miles for the ocean ships, their cargo deliveries compare as follows:

$V/\sqrt{L}$	*Time Spent at Sea*	*Time Spent in Harbor*	*Total Time*	*Round Trips per Year*	*Cargo Delivered*
0.72	38.75	11.85	50.60	6.52	109,125
0.85	31.30	8.50	39.80	8.30	102,400
1.11	23.20	2.64	25.84	12.75	55,500

As in the coaster estimate, the return trip is made with only one-half of the full cargo weight in each case. But, quite the reverse from the coasters, the slowest ocean ship delivers the most cargo per year, and the fastest is quite out of the reckoning even allowing her less than three days in harbor—evidently too short a time. The harbor-time formula, page 199, seems to favor the long ship with the low cargo weight but works well in the case of the average cargo carrier. It might also be questioned if harbor facilities are such that 6,650 tons of oil and water can be bunkered in less than three days.

To complete this analysis, an estimate has been made of the cargo delivery of the vessel with a speed-length ratio of 0.63. Although her dead weight would amount to 14,250 tons, she could only deliver 106,050 tons per year, about 3% less than the slowest ship in the table above. The statement in the text that a speed-length ratio of 0.72 is most economic for a cargo vessel except for very large ships, is thus borne out here.

So far, only cargo delivery has been considered as it is most important in wartime, and sometimes in peace too, as, for instance, when food must be shipped quickly to starving multitudes. But no private shipowner would ever build or acquire vessels except for the possible profit derived therefrom—be it peace or war. Now, profit is a very changeable thing, and is what remains to the ship owner after all expenses have been deducted from the gross income. The latter depends on the cargo rate per ton, and on the amount of cargo delivered in a given time. In a paper read in 1920 before *The Institution of Engineers and Shipbuilders* in Scotland, the writer showed how fitfully the gross earnings varied from year to year for British and for Norwegian shipowners. Fortunately, the fuel expenses varied in much the same way, except the seamen's wages which the labor unions tried to fix at a high level whatever the gross earnings of the shipowners might be.

The aforesaid paper shows also that profit-and-loss accounting is mighty intricate work, quite outside the pale of this book, but it should

be clear to all concerned that first costs and fuel expenses are main items in the account. Although first costs are usually quoted per deadweight ton, actually such costs are dependent on the weight of the finished ship *in toto* but without fuel, stores, etc. The cost of all machinery on board, per ton, is about two times as high as the cost of the hull per ton. The weight of the machinery of the slowest ocean ship in the table, is only about one-half and one-fourth of the machinery weights of the other two vessels that thus can be dismissed from consideration. In fact, the ship with a speed-length ratio of 0.63 would cost the least per ton of cargo delivered, and thus earn the most profit—were it not for the fact that the cargo rates increase with the speed of the vessel, although not in exact proportion. Besides, in time of dearth of commodities for shipments, the cargo always goes to the fastest ship. This fact may be accentuated in the future on account of competition with airships, and it might be a moot question if a speed-length ratio of 0.72 always makes cargo ships most profitable to the owners of ocean-going vessels.

As regards coasters, the question of the most favorable speed-length ratio is not so easily answered. Although the fastest ship in the table on page 201 can deliver most cargo per year, she evidently costs more to build and run, but would also command the highest cargo rates, at least when cargo offerings are plentiful. The most favorable speed-length ratio is one that offers the highest return for capital invested. Reduced to its simplest form,

$$\text{Return } P.A. = \frac{\text{cargo weight} \times \text{rate} - \text{charges}}{\text{invested capital}}$$

The cargo rate per ton can either be constant for all speeds, or can be proportional to the speed of the ship. The charges, except fuel and lubricating oils, are estimated as a percentage of the invested capital. Assuming a minimum cargo rate of $3.75, the price of oil $20. per long ton, and the charges as 28% of the invested capital, the following table shows the result.

Returns on Invested Capital

				Remarks
Speed-length	0.72	0.85	1.11	
Cargo weight *P.A.*	47,500	51,750	58,000	Page 201
Rate per ton	3.75	4.65	6.28	Prop. to speed

				Remarks
Product	178,200	240,500	364,000	
Invested capital	483,100	570,300	859,200	1939 prices
Charges *P.A.*	135,200	159,500	240,200	28%
Fuel cost *P.A.*	16,650	31,700	80,200	$20. per ton
Total charges	151,850	191,200	320,400	
Net profit *P.A.*	26,350	49,300	43,600	
Return percent.	5.45	8.63	5.00	

A speed-length ratio of 0.85 thus offers the highest return per annum, equal to 8.63%. If the rate were constant for all three ships, the return would be 5.45, 0.52, and a big loss. The cargo rate for no return is $3.20, $3.70, and $5.52 per ton.

It might, perhaps, be argued in this and similar cases for coasters, that some intermediate speed-length ratio, for example 0.76, might offer still higher returns. The higher wave resistance per ton of displacement, resulting in a much larger engine, more fuel weight and less cargo weight, as well as the increases in capital invested and charges, make all intermediate speed-length ratios less profitable to the shipowner.

Conclusion. For coasters and smaller ocean ships, a speed-length ratio of 0.85 is most favorable to the shipowner according to this analysis. In case of ocean ships, however, the most favorable speed-length ratio is 0.72, except for vessels of the largest class over 20,000 tons dead weight, when 0.63 would be best. These speed-length ratios should be used for vessels carrying both passengers and cargo, but for pure passenger ships that are always subsidized, a maximum ratio of 1.11 could be used, equivalent to 33.3 knots for a 900-ft. ship.

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